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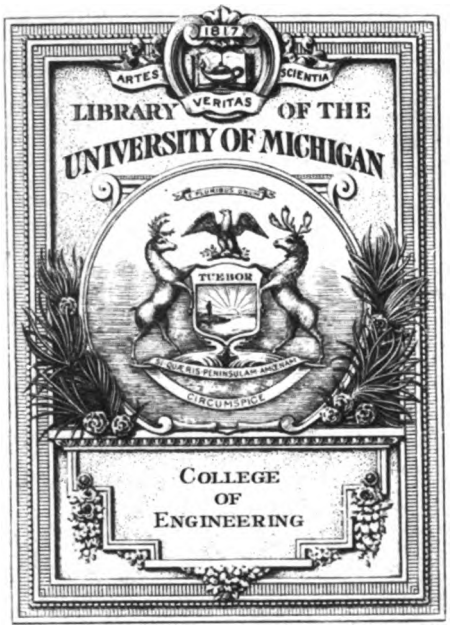


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Mountain, W. C.
Mulherion, G. F.
Mullan, F. C.

Napier, James, M.A., F.R.S.E
Neal, William George
Neilson, John
Newman, A. R.
Nicholson, C. E.
Niven, O. B., Com.
Niven, W. D., C.B., F.R.S.
Noble, Sir Andrew, K.C.B., F.R.S. (Vice-President)
Noel, Sir Gerard H., Rear-Admiral, K.C.M.G. (Assoc. Mem. Council)
Northbrook, The Right Hon. the Earl of, G.C.S.I. (Vice-President)

O'Connor, C.
Ormston, C. W.
Osbourne, James.
Ouless, Philip D.
Ouspensky, I. P., Captain, Naval Attaché to the Imperial Russian Embassy

Panton, Charles E.
Park, J. Smith, Major V.D.
Parker, J. Henry
Parsons, P. R.
Pascoe, J. R.
Pearce, Sir William G., Bart., M.A., LL.B.
Peat, James Barclay
Pendred, Loughnan St. Lawrence
Percival, J. W.
Perkins, James
Perkins, T.
Petersen, Otto L.
Petersen, William
Petri, Charles
Pharo, Jacob C.
Philip, G. N.
Pile, Charles H.
Pinkney, Thomas
Pitt, F. J., Captain, R.N.
Plater, H. R. F.
Pratt, G. H. F.
Prendiville, John Edward
Preston, F. G. Panizzi
Preston, H. W.
Putnam, Thos.

Ravier, Sylvain Louis
Rawson, Robert, Hon. Mem. Manchester Literary and Philosophical Society
Read, William
Reid, William
Reincke, R. Leopold H.
Renton, A. H.
Renwick, George
Reynolds, Edward Parker
Richardson, J. H.
Richardson, Sir Thomas, M.P.
Ridley, John H.
Riley, James (Assoc. Mem. Council)
Ritson, Arthur
Robson, George
Rognetta, F. B., Colonel
Rosenthal, J. H.
Roseti, G. S., Captain, Argentine Navy
Ross, A. J. C.
Rosse, The Right Hon. the Earl of

Rothe, H. P., Commodore, Danish Royal Navy
Rowley, Charles J., Admiral
Rugg, C. H.
Ruhm, T. S.
Runciman, W.
Runciman, Walter, Jun., M.P.
Russell, A. G.

Saborido, L., Lieut., Argentine Navy
Sachs, E. O.
Samuelson, The Right Hon. Sir Bernhard, Bart., F.R.S.
Sanders, Ludwig
Sankey, M. H. P. Rial, Captain, R.E. (retired)
Saunders, James Ebenezer
Scantlebury, Vincent John
Schlutow, Albert
Scholefield, A.
Schwanhausser, W.
Scott, Colin W.
Scott, Ernest
Scott, George J.
Scott, James Henry
Seaman, Charles J.
Semler, Johannes
Serena, Arthur
Seymour, Sir Edward Hobart, G.C.B., Vice-Admiral
Shillington, Robert W.
Simpson, J. B.
Singer, F. M.
Sloan, W.
Smith, Benjamin Leigh
Smith, Clarence Dalrymple
Smith, Eustace.
Smith, F. Henry
Smith, R. Knightly
Smith, S. Farrar
Smith, Thomas Knightly.
Smith-Carington, H. H.
Spence, G. O.
Spencer, Right Hon. Earl, K.G. (Vice-President)
Spencer, John W.
Spooner, G. P.
Starnes, J. S.
Stephens, Daniel



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Stephens, R. E.
Stephens, Thomas Walls
Stewart, Andrew
Stewart, J. G.
Stewart, Walter J. L.
Stileman, F.
Stoker, R. B.
Sutherland, Sir Thomas, G.C.M.G., M.P.
Swinburne, M. W.
Swinton, A. A. C.
Sztranyavszky, L. von, Captain, Naval Attaché to the I. and R. Austro-Hungarian Embassy

Taite, John Charles
Tamplin, F. A.
Tamplin, T. Ward
Tate, R.
Tawse, James
Taylor, George W.
Taylor, W. Isaac
Temperley, Joseph
Temperley, J. R.
Terry, Stephen H.
Thomson, Andrew
Thompson, O. S.
Thompson, Stephen
Thornycroft, J. E.
Thurston, Robert H., Professor
Tiedeman, Carl
Todd, J. Stanley
Torrey, C. F.
Trotter, T., Captain

Uribe, Luis, Rear-Admiral, Chilian Navy
Utley, T.

Van Raalte, J.
Vavasseur, Josiah
Vickers, A.

Wade, W. S.
Wait, A. M.
Walker, W., Colonel, V.D., J.P.
Wallis, R.
Watkinson, W. H., Professor
Watson, H. Burnett
Watson, Henry James
Watson, W.
Watson, Sir William
Watson-Armstrong, W. A.
Watt, John
Watts, Fenwick Shadforth
Webster, M. P.
Westmacott, Percy G. B.
Wharton, Sir William J. L., K.C.B., F.R.S., Rear-Admiral
White, J. Bell
White, Robert
Whitehead, Henry
Wiegand, H.
Willard, C. D.
Williams, R. W.
Williamson, R. H.
Wills, W.
Wilmot, S. Eardley, Rear-Admiral
Wilson, Sir Alexander, Bart.
Wilson, A. G.
Wilson, A. K., Rear-Admiral, V.C., C.B.
Wilson, W. G.
Wingfield, G. T., Com. R.N.
Wise, William Lloyd
Woodall, J. W.
Worsdell, Wilson
Wylie, R.
Wylie, William

Yamamoto, N.
Younger, Robert L.

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# OBJECTS OF THE INSTITUTION.

The objects of the Institution of Naval Architects—which was established to promote the Improvement of Ships, and of all that specially appertains to them—are comprised under three heads :—

First, the bringing together of those results of experience which so many shipbuilders, marine engineers, naval officers, yachtsmen, and others acquire, independently of each other, in various parts of the country, and which, though almost valueless when unconnected, doubtless tend much to improve our Navies when brought together in the printed Transactions of an Institution.

Secondly, the carrying out, by the collective agency of the Institution, of such experimental and other inquiries as may be deemed essential to the promotion of the science and art of shipbuilding, but are of too great magnitude for private persons to undertake individually.

Thirdly, the examination of new inventions, and the investigation of those professional questions which often arise, and were left undecided before the establishment of this Institution, because no public body to which professional reference could be made then existed.

# BYE-LAWS AND REGULATIONS.

## CONSTITUTION.

1. The Institution of Naval Architects shall consist of four classes, viz., Members, Associate Members, Associates, and Honorary Members.

2. *Members.*—The Class of Members shall comprise every person who on the 22nd March, 1899, was on the Register of Members, and every Naval Architect or Marine Engineer thereafter elected or transferred into the class of Members.

The Class of Associate Members shall comprise every Naval Architect and Marine Engineer who, under the provisions of Rules 43 and 44, may be elected into this class.

3. *Associates.*—The Class of Associates shall consist of persons who are qualified either by profession or occupation, or by scientific or other attainments, to discuss with Naval Architects the qualities of a ship, or the construction, manufacture, or arrangement of some part or parts of a ship or her equipment.

4. *Honorary Members.*—The Class of Honorary Members shall consist of persons upon whom the Council may see fit to confer an honorary distinction.

## ELECTION AND DUTIES OF OFFICERS.

5. The Officers of the Institution shall consist of a President, Vice-Presidents, Members of Council, Associate Members of Council (not exceeding in number one-third the number of Members of Council) a Treasurer, two Auditors of Accounts, and a Secretary or Secretaries.

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6. A General Meeting of the Members, Associate Members, and Associates of the Institution shall be held annually before Easter in each year; and at this Annual General Meeting the Members of Council, Associate Members of Council, Treasurer, and Auditors for the ensuing year shall be elected.

7. At the Annual General Meeting Members and Associate Members only shall vote in the Election of Members of Council, and both Associates, Members, and Associate Members in the election of Associate Members of Council, the Treasurer, and the Auditors.

8. *President.*—Both Members and Associates of the Institution shall be eligible for election as President. The President shall preside over all meetings of the Institution, and of Officers of the Institution, at which he is present, and shall regulate and keep order in the proceedings.

9. *Vice-Presidents.*—Both Members and Associates of the Institution shall be eligible for election as Vice-Presidents. In the absence of the President, one of the Vice-Presidents shall preside at the General Meetings of the Institution, and shall regulate and keep order in the proceedings.

10. In case of the absence of the President and of all the Vice-Presidents, the Meeting may elect any Member of Council or Associate Member of Council, and in case of their absence any Member present to preside.

11. The Chairman at any Meeting of the Council of the Institution, when the votes of the Meeting, including his own, are equally divided, shall be entitled to give a casting vote.

12. Persons holding the office of Vice-President shall at all times be entitled to sit and vote with the Council.

13. *Past Presidents and Vice-Presidents.*—All Members who have held the posts of President and Vice-President shall, while their connection with the Institution as Members lasts, be entitled to sit and vote with the Members of Council.

14. *Members of Council.*—Members only shall be eligible for election as Members of Council at the Annual General Meeting.

15. *Associate Members of Council.*—Associates only shall be eligible for election as Associate Members of Council at the Annual General Meeting.

16. The Direction and Management of the Institution shall be vested in the Council for the time being, the Associate Members voting with the Members of Council in all cases, except in the decision of questions directly affecting the forms of ships and the construction of their hulls.

17. The Council shall meet as often as the business of the Institution requires, and at every Meeting five Members of the Council shall form a *quorum.*

18. The Council may appoint Committees to report to them upon special subjects.

19. All questions shall be decided in the Council by vote; but at the desire, expressed in writing, of any four Members or Associate Members present, the determination of any subject shall be postponed to the succeeding meeting of the Council.

20. An annual statement of the funds of the Institution, and of the receipts and payments of the past year, shall be made under the direction of the Council, and, after having been verified and signed by the Auditors, shall be laid before the Annual General Meeting.

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21. The Council shall draw up an Annual Report on the state of the Institution, which shall be read at the Annual General Meeting.

22. It shall be the duty of the Council to adopt every possible means of advancing the Institution, to provide for properly conducting its business in all cases of emergency, such as the death or resignation of Officers, and to arrange for the publication of the Papers read at the Meetings, or of such documents as may be calculated to advance the objects of the Institution.

23. *Treasurer.*—Only Bankers, or Members of Council, or persons who have been Members of Council and are still Members of the Institution, shall be eligible for election as Treasurer.

24. *Trustees.*—There shall be four Trustees, two of whom shall be the President and Treasurer of the Institution for the time being. The remaining two shall be appointed by, and hold office at the pleasure of the Council. In the names of these trustees, under the direction of the Council of the Institution, all securities shall be taken and investments made, the whole of such property being, notwithstanding, subject to the disposition of the Council, and the order of the Council in writing, signed by the Chairman of the Meeting and countersigned by the Secretary, shall be obligatory upon and full authority for the Trustees.

25. *Auditors.*—All Members, Associate Members, and Associates of the Institution shall be eligible for election as Auditors.

26. The Auditors shall have access at all reasonable times to the Accounts of the pecuniary transactions of the Institution; and they shall examine and sign the annual statement of the Accounts before it is submitted by the Council to the Annual General Meeting.

27. *Secretary.*—The Secretary or Secretaries shall be elected by the Council, and shall be removable at the will of the Council, after due notice given. The salary of the Secretary or Secretaries shall be fixed by the Council.

28. It shall be the duty of the Secretary, under the direction of the Council, to conduct the correspondence of the Institution; to attend all Meetings of the Institution and of the Council; to take Minutes of the proceedings of such Meetings; to read the Minutes of the preceding Meeting; to announce donations made to the Institution; to superintend the publication of such Papers as the Council may direct; to have charge of the library, museum, and offices of the Institution; and to direct the collection of subscriptions and the preparation of accounts. He shall also engage, and be responsible for, all persons employed under him, and generally conduct the ordinary business of the Institution.

29. In each year six Ordinary and two Associate Members of Council shall retire, unless before the date of drawing up the Balloting Lists for the election of the Council any Members of the Council shall have died or resigned, in which case only so many members shall retire as shall be necessary in order to make up the number to six Ordinary and two Associate Members of Council, subject always to the provisions of Rule 36. The Members who shall retire in each year shall be those who have served longest on the Council from the date of the last election, and in the event of there being several Members who have served an equal time on the Council, the order of retirement amongst these shall be alphabetical. The retiring Members shall be eligible for re-election.

30. In January of each year the Council shall meet and prepare Lists for the election of the Council for the ensuing year. These Lists shall be as follow, namely :—

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*1st.* A List of the names of the President, Vice-Presidents, and Treasurer for the ensuing year to be submitted at the Annual General Meeting, for their election in a body.

*2nd.* A list of candidates to fill any vacancies in the list of Vice-Presidents that it may be intended to fill up from among the professional Members of the Institution.

*3rd.* Lists for the election of the Ordinary Members and Associate Members of Council.

31. No addition shall be made to the list of Vice-Presidents until, by death or resignation, their number shall have been reduced to below twenty-four, after which their numbers shall be raised to and preserved at twenty-four. The vacancies are to be filled up in such a manner that not less than one-half nor more than two-thirds of the total list of Vice-Presidents shall be professional Members of the Institution. Provided always that the Council shall be at liberty, should special circumstances arise before the numbers shall have been reduced below twenty-four, to provide for the election of one Member and one Associate of the Institution as Vice-Presidents.

32. When any vacancy occurs in the list of Vice-Presidents which it is intended to fill by the election of a professional Member of the Institution, the election shall be by voting papers issued to all Members and Associate Members of the Institution. The candidates to fill the vacancy shall be selected by the Council in the month of January from among the existing or past Members of Council. The number of candidates shall not be less than two for each vacancy, but shall not otherwise be limited. The voting papers shall be issued to the Members and Associate Members at the same time as the voting papers for the election of Members of Council, and shall be subject to the same regulations and scrutiny as these latter, as provided for by Rules 37, 38, 39, 40.

33. After having been once elected by voting papers, the Vice-Presidents will be subject to re-election every year in a body at the Annual Meetings.

34. When any vacancy occurs in the list of Vice-Presidents which it is intended to fill by the election of an Associate of the Institution, the nomination to fill the vacancy shall be made by the Council, and the candidate nominated shall be included in the list of Vice-Presidents submitted at the Annual Meetings for election in a body.

35. No addition shall be made to the total number of Ordinary Members of the Council until, by death of resignation, their numbers shall have been reduced below twenty-four, after which their numbers shall be raised to and preserved at twenty-four. And no addition shall be made to the Associate Members of Council until, by death or resignation, their numbers shall have been reduced below eight, after which their numbers shall be raised to and preserved at eight, always exclusive of the President, Vice-Presidents, and Treasurer.

36. At the date of issuing the Syllabus of the Annual General Meetings in each year, the Lists proposed by the Council for the election of Members to fill the vacancies in the Ordinary Council for the ensuing year shall be printed, and sent to all Members and Associate Members to serve as Balloting Lists. These Lists shall contain, first, the names of the retiring Ordinary Members of Council at the time of the preparation of the Balloting List, together with as many new names of Members of the Institution as shall be needed to bring the number up to twice the number of vacancies, and the whole of these names shall be printed in alphabetical order. Secondly, the names of the retiring Associate Members of Council at the time of the preparation of the Balloting List, together with as many new names of Associates of the Institution as shall be needed to bring the number up to twice the number of vacancies, and these names also shall be printed in

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alphabetical order. From these Lists the vacancies in the Council shall be filled up. Every Member and Associate Member shall be at liberty to vote for as many names on each of the Lists as there are vacancies to be filled, but not for more.

37. A similar Balloting List (in which, however, the names of the Ordinary Members of Council proposed for election shall not be included) shall be printed and sent to all Associates of the Institution, to serve as a Balloting List for Associates, from which the voting for Associate Members of Council shall be taken. Every Associate shall be at liberty to vote for as many names on that List as there are vacancies to be filled, but not for more.

38. The Balloting Lists may be sent by post or otherwise to the Secretary, so as to reach him before the day and hour named for the Annual General Meeting, or they may be personally presented by the Members, Associate Members, and Associates at the opening of the Annual General Meeting.

39. At the opening of the Annual General Meeting the order of business shall be :—

(1) To read and consider the Reports of the Council and Treasurer.

(2) To read the List of Officers and Nomination for Council for the ensuing year, proposed by the Council.

(3) The Chairman shall next put to the Meeting the List containing the names of the President, Vice-Presidents, and Treasurer for election for the ensuing year.

(4) The Chairman shall then nominate two Scrutineers (of whom one only shall be a Member of the existing or proposed Council), and shall hand to them the Ballot Boxes containing the Voting Papers for the Ordinary Members of Council and Associate Members of Council ; and

(5) The Scrutineers shall receive all Ballot Papers which may have reached the Secretary, and all others which may be presented by Members, Associate Members, or Associates at the Meeting. The Scrutineers shall then retire and verify the Lists, and count the votes ; and shall, not later than the following day, report to the Chairman the names which have obtained the greatest number of votes, subject to the conditions of the Ballot. The Chairman shall then read the List presented by the Scrutineers, and shall declare the gentlemen named in the List to be duly elected, provided always that the List does not contain more names than there are vacancies to be filled. If, in consequence of two or more of the candidates receiving an equal number of votes, the List shall contain more names than there are vacancies, the Council shall, at their next meeting, decide which of these candidates shall be elected.

(6) After the Ballot shall have been taken, and the Scrutineers have retired, the Meeting will proceed to the other business before it.

40. The new Council and Officers shall take office immediately after the close of the Annual General Meeting.

41. In the event of any vacancy occurring in the offices of either President or Treasurer after the date of the Annual Election in any year, the Council shall have power to elect a new President or Treasurer as the case may be, who shall hold office till the conclusion of the next Annual General (Spring) Meeting of the Institution.

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## DESIGNATION OF MEMBERS AND ASSOCIATES.

42. Any Member, Associate Member, Associate, or Honorary Member, having occasion to designate himself as belonging to the Institution, shall state the class to which he belongs according to the following abbreviated forms, viz., M.I.N.A.; A.M.I.N.A.; Associate I.N.A.; Hon. Mem. I.N.A.

## ELECTION OF MEMBERS, ASSOCIATE MEMBERS, AND ASSOCIATES.

43. *Admission of Members.*—Every Candidate for admission or for transference into the Class of Member shall be more than twenty-five years of age. Every Candidate for admission or for transfer into the Class of Associate Member shall be more than twenty-two years of age. Every Candidate in either of these classes shall comply with the following Regulations:—

He shall submit to the Council a statement showing that he has—

(*a*) Served an apprenticeship, or pupilage, in shipbuilding or marine engineering for at least five years, or

(*b*) Undergone a mixed training of at least four years, partly in shipbuilding or marine engine works, and partly in a School of Naval Architecture or Marine Engineering recognised by the Council of the Institution.

He shall further show that, after completing his training, as set forth under headings (*a*) or (*b*), he has been professionally employed in some public or private shipbuilding yard, or marine engine works, or registration society, or as professional superintendent of a steamship company, and shall give full particulars of the nature and duration of such employment.

In the event of the Candidate not having complied with any of the conditions under headings (*a*) or (*b*), he shall show—

(*c*) That he has been regularly educated by apprenticeship, or pupilage, or in a technical school, as a civil or mechanical engineer, and that he has been subsequently *professionally* employed for at least seven years in some public or private shipbuilding yard or marine engine works.

This statement shall be signed by at least three Members of the Institution, whose signatures shall certify their personal knowledge of the Candidate, and approval of his statement.

44. These preliminary conditions being satisfied, the Council shall then decide whether the education, the practical experience, and the professional attainments of the Candidate are such as entitle him to be recommended by the Council for election to the Institution, and, if so, whether as a Member, or as an Associate Member. The proposal of the Candidate for admission to the class decided on by the Council shall be submitted to the Members of the Institution (who shall have access to the applicant's statement), at an Ordinary Meeting of the Institution, for them to vote upon, the voting to be by ballot, should a ballot be demanded.

45. *Admission of Associates.*—Candidates for Associateship shall submit to the Council a proposal for their admission, setting forth therein a statement of their claims to be admitted as Associates. This proposal shall be signed by at least one Member, Associate Member, or Associate of the Institution. Their proposal, if approved by the Council, shall be submitted by them at an Ordinary Meeting of the Institution, for the Members, Associate Members, and Associates jointly to vote upon, the voting to be by ballot, should a ballot be demanded.

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46. The proportion of votes for deciding the election of Members, Associate Members, and Associates shall be at least four-fifths of the numbers recorded.

## SUBSCRIPTIONS.

47. Each Member, Associate Member, and Associate shall pay an Entrance Fee of two guineas, and an Annual Subscription of two guineas in advance; the first Subscription being payable on his election, and all future ones on the 1st day of January of each year. Any Member or Associate withdrawing from the Institution after that date is still liable for the amount of Subscription due on that day.

48. Any Member, Associate Member, or Associate may compound for his Annual Subscription, for life, by a single payment of not less than thirty guineas.

49. No person's name shall be entered on the Roll as Member, Associate Member, or Associate of the Institution nor possess the privileges of Membership (except it be on the honorary list) until he shall have paid his first subscription or the life composition, and if the payment be delayed for more than twelve months from the date of his election, the same shall be void unless the Council otherwise direct.

50. The Secretary shall at the close of every year notify to all Members, Associate Members, and Associates whose subscription for that year shall not have been paid, that it will be his duty to report accordingly to the Council, and he shall at the same time furnish the person whose subscription is in arrear with copies of this and the two following Rules.

51. The Secretary shall before Easter in every year lay before the Council a list of all Members, Associate Members, and Associates whose subscriptions for the two previous years shall be still unpaid, and unless the Council shall otherwise direct, the names of those in arrear shall be expunged from the Roll of Members, Associate Members, and Associates, and shall not be replaced without re-election in due form. Provided always that the Council shall at any time within two years therefrom have power to dispense with such re-election, and to restore the name to the Roll upon payment of all subscriptions then due, and upon cause being shown to the satisfaction of the Council why such subscriptions were not previously paid.

52. Nothing herein contained shall prejudice the right of the Institution to the legal recovering of all arrears of subscriptions up to the date of striking the name off the Roll.

53. In case the Council shall be of opinion that any Member, who has been long distinguished in his professional career, from ill-health, advanced age, or other sufficient causes, should not be called upon to continue his annual subscription, they may remit it. Also they may remit any arrears which are due from an individual, or may accept a collection of books, or drawings, or models, or other such contribution as, in their opinion, under the circumstances of the case, may entitle the person to be enrolled as a Life Subscriber, or to enable him to resume his former rank in the Institution which may have been in abeyance from any particular causes. These cases must be considered and reported upon by a Sub-Committee named for the purpose.

54. In case the expulsion of any individual shall be judged expedient by ten or more Members, and they think fit to draw up and sign a proposal requiring such expulsion, the same being delivered to the Secretary shall be by him laid before the Council. If the Council, after due inquiry, do not find reason to concur in the proposal, no entry thereof shall be made in any Minutes, nor shall any public discussion thereon be permitted; but if the Council do find good reason for the proposed expulsion they shall direct the Secretary to address a letter to the person proposed to be expelled, advising him to withdraw from the Institution. If that advice be followed, no entry on the Minutes nor any public discussion on the subject



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shall be permitted; but if that advice be not followed, nor a satisfactory explanation given, the Council shall call a Special General Meeting of Members, Associate Members, and Associates, for the purpose of deciding on the question of expulsion; and if two-thirds of the persons present at such Special General Meeting, providing the number so present be not less than thirty, vote that such individual be expelled, the Chairman of that Meeting shall declare such expulsion accordingly, and the Secretary shall communicate the same to the individual.

## MEETINGS.

55. Meetings for the Reading of Papers shall be held as frequently, and at such times, as the Council may determine.

## TRANSACTIONS.

56. The *Transactions* of the Institution, including the Papers read at the Ordinary Meetings, and Reports of the Discussions by which they are followed, shall be edited by the Secretary, and printed under the direction of the Council.

57. A copy of each Volume of *Transactions* shall be sent free to every Member, Associate Member, and Associate.

58. The Secretary, under the direction of the Council, may dispose of the surplus stock of *Transactions* which have been published more than three years, at a price of not less than One Guinea a volume, provided a sufficient number remain on hand to supply the probable demand of New Members and Associates to complete their sets by the purchase of the back Volumes.

## CHANGE OF ADDRESS.

59. Members, Associate Members, and Associates are particularly requested to communicate to the Secretary any change of address.

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# PROCEEDINGS IN LONDON.

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SPRING MEETINGS OF THE FORTY-SECOND SESSION

OF THE

# INSTITUTION OF NAVAL ARCHITECTS.

MARCH 27, 28, AND 29, 1901.

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# INTRODUCTORY PROCEEDINGS.

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THE Spring Meetings of this, the Forty-second Session of the Institution of Naval Architects, were held on March 27, 28, and 29, 1901, in the Hall of the Society of Arts, John-street, Adelphi, W.C.

The opening meeting was presided over by the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President of the Institution, who commenced the proceedings by calling on the Secretary, Mr. George Holmes, to read the Report of the Council, which was as follows:—

## ANNUAL REPORT OF THE COUNCIL, 1901.

The first duty of the Council in the present year is to refer to the great loss which the nation has sustained by the death of our late beloved Sovereign Queen Victoria. The grief which has been caused by this melancholy event has been fully shared by the Members of this Institution, who remember, with pride and gratitude, the direct encouragement and kindness which the Institution received from Her late Majesty in the year of the Diamond Jubilee.

The Council, on behalf of the Institution, have forwarded a dutiful and loyal address of condolence to His Majesty the King, who as Prince of Wales conferred upon the Institution the distinction of becoming one of its honorary members. The Council have also tendered their congratulations to His Majesty on his accession to the Throne.

The Council have likewise to allude with much regret to another melancholy death in the Royal Family which directly affects the Institution. H.R.H. Duke Alfred of Saxe-Coburg-Gotha, by his decease last August, has rendered vacant the Vice-Presidency which he held as a mark of the great interest which he felt in the Institution. The Council have, however, much satisfaction in announcing that H.R.H. the Duke of Cornwall and York, now Heir Apparent to the Throne, has consented to fill the vacancy thus created, and has thereby given a proof of the value which he attaches to our labours.

The Council have also to report to the Members that they have lost the valuable services of their late President, the Earl of Hopetoun. Lord Hopetoun filled the chair of this Institution during one of the most memorable periods of its history. The success which attended the meetings of the Institution in Germany in 1896, and the meetings of the International Congress of Naval Architects and Marine Engineers convened by this Institution in the following year, were due in the highest degree to his devotion to our interests, and to his great tact and personal influence.

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It was inevitable that a man of Lord Hopetoun's great talents and high personal character should be chosen to fill one of the highest offices under the Crown, and, when the Australian Commonwealth was created, our late President was selected by Queen Victoria for the important post of its first Governor-General. While we deeply deplore his loss, we congratulate our Australian fellow-subjects upon having acquired the services of such a distinguished statesman, and the best wishes of the Members of this Institution will follow him to his new sphere of duties. Before his departure for Australia the Council, on behalf of the Institution, presented Lord Hopetoun with an Address, the text of which is appended to this Report.

Acting under the provisions of Rule 41, the Council have elected the Right Hon. the Earl of Glasgow, G.C.M.G., as President of the Institution till the close of the annual general meeting, and have now much pleasure in proposing him for election by the Members. Lord Glasgow has occupied the important post of Governor of New Zealand. He is a retired captain in H.M. Navy, and is well qualified by his knowledge of naval matters, and by his past experience, for the office of President.

The Council have much pleasure in reporting that the finances of the Institution continue to be in a satisfactory condition. The receipts under the heading of entrance fees, when compared with the year 1899, show a slight diminution, due to the fact that no Summer Meeting was held last year.

As was announced at the previous Spring Meetings, the Council have received and accepted a very kind invitation from the Lord Provost, Magistrates, and Corporation of Glasgow, to visit that city during the coming summer. The Meetings have now been definitely fixed to take place on Tuesday, June 25, and the three following days. An important International Exhibition will be held in Glasgow during the summer, and will render the Meetings more than usually attractive.

An International Engineering Congress will also be held in Glasgow early in September. At the request of the General Committee of the Congress the Council have undertaken the management of the Section relating to Naval Architecture and Marine Engineering.

Last July an important International Congress of Naval Architecture was held in Paris under the Presidency of our esteemed Honorary Member, Monsieur L. de Bussy. As all the Members of this Institution received invitations to be present, and as the Council was most anxious to further the success of this Congress, it was deemed inadvisable to hold a Summer Meeting last year.

The Société des Ingénieurs Civils de France invited the Council to send representatives to be present at a series of receptions and fêtes given by the Société last June in honour of the technical and scientific societies of all nations, on the occasion of the holding of the International Exhibition in Paris.

The Members of the Institution who attended the Congress, and the official delegates who were deputed to represent the Council, both at the Congress and at the series of receptions mentioned above, were received with the hospitality and graceful courtesy which we have always experienced from the French Government and from our colleagues in France. The warm thanks of the Institution have been conveyed to their French hosts for the kindness they received on these memorable occasions.

Last year the Committee which organised a testimonial in honour of Mr. B. Martell, on the occasion of his retirement from the position at Lloyd's Register of British and Foreign Shipping, which he had held with so much distinction for many years, acting in accordance with the wishes

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of Mr. Martell, offered to place at the disposal of the Institution the sum which had been contributed for the purpose of endowing a Scholarship in Naval Architecture to be associated in perpetuity with Mr. Martell's name. The Council gratefully accepted this fund, and have drawn up regulations for the administration of the Scholarship, which are appended to this Report.

The Council have had much pleasure in awarding the Gold Medal for the year 1900 to Professor G. Bryan, F.R.S., for his very valuable paper, "The Action of Bilge Keels."

In addition to the death of H.R.H. Duke Alfred of Saxe-Coburg-Gotha, alluded to above, the Council have also to deplore the death of another of its most distinguished Vice-Presidents, the Lord Armstrong, C.B., whose loss to the profession and to the Institution will be most severely felt. The Council also deeply regret to have to report the death of Mr. C. Ferdinand Laeisz, of Hamburg, an Associate of the Institution. The part which Mr. Laeisz took in the organisation of the German Meeting in 1896, and the active interest which he took in the work of the Institution, will always be gratefully remembered. His death is a severe loss both to his country and to this Institution.

### Text of Address Presented to the Right Hon. the Earl of Hopetoun, K.G., G.C.M.G., P.C., President of the Institution of Naval Architects.

We, the Vice-Presidents, Members of Council, and Members of all denominations of the Institution of Naval Architects, desire to offer your Lordship our sincere congratulations on your selection by the Queen as first Governor-General of the Australian Commonwealth.

We know that your Lordship will bring to the discharge of the duties of this office the high qualities that have marked the whole of your public career, and that the Empire, and especially our Australian fellow-subjects, will benefit greatly by your appointment. At the same time we cannot but feel that the public gain is our loss. We deeply regret that the acceptance of your new post of necessity involves your resignation of the Presidency of this Institution. During the five years that you have held this office you have conducted the affairs of the Institution with the highest wisdom and ability, and have won the respect and affection of all its Members. We recognise with gratitude your unfailing courtesy, and the spirit of fairness and impartiality with which you have presided over us. We feel that upon all occasions you have done everything that was humanly possible to advance our interests.

We thank you for your untiring efforts. We wish you and Lady Hopetoun all happiness and prosperity in your new career, and we look forward with hopefulness to the day when we shall see you once more in our midst.

Signed on behalf of the Institution of Naval Architects,

John Dalrymple-Hay,
E. J. Reed,
Nathl. Barnaby, } *Vice-Presidents.*

George Holmes, *Secretary.*

*October 1st*, 1900.

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### Recommendations of the Martell Scholarship Committee.

A Scholarship designated "The Martell Scholarship in Naval Architecture" will be granted by the Council of the Institution of Naval Architects who have been appointed administrators of the Fund available for providing the Scholarship. The Scholarship is of the annual value of about £50, and, subject to the conditions named below, is tenable for three years.

(1) Candidates for the Scholarship must forward a written application to the Secretary of the Institution of Naval Architects, 5, Adelphi-terrace, London, W.C., in time to reach him not later than March 1 immediately preceding the holding of the examination referred to in Clause 3.

(2) Candidates are not to be less than eighteen years of age nor more than twenty-one years of age on March 1 of the year of examination, and must, at that date, have been continuously employed for at least two years upon Naval Architecture or Marine Engineering. Only candidates who shall obtain certificates in the examination held by the Science and Art Departments in the following subjects shall be eligible for the Scholarship.

(3) The subjects of examination to be as follows :—

I. Naval Architecture.
II. Practical Mathematics.
III. Theoretical Mechanics.

Any stage may be taken at these examinations, but the successful candidate must obtain at least a first-class pass in the advanced stage, or a pass in some higher stage of Naval Architecture.

(4) No candidate will be appointed to the Scholarship till he has produced a certificate of good character, and of time of service from his employers, and a certificate of birth.

(5) Each candidate appointed to the Scholarship will be required to undergo a three years' course of study in such University, or College, combined, as far as possible, with a practical training in a shipyard or marine engine works, during the college vacations, as he may propose and the Council shall approve.

(6) A report is to be sent in to the Council at the end of each session of study by the holder of the Scholarship of the progress made, and the results of examinations attended. If the progress made should, in the opinion of the Council, be unsatisfactory, or if the holder of the Scholarship should, in the opinion of the Council, cease to comply with any of the conditions of tenure, the Council will determine the Scholarship and fill the vacancy thus arising.

(7) All communications with respect to this Scholarship should be addressed to the Secretary of the Institution of Naval Architects, 5, Adelphi-terrace, London, W.C., and should be endorsed on the upper left-hand corner of the envelope "Martell Scholarship."

(8) The Trustees of the Institution for the time being to be the Trustees of the Scholarship Fund.

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## Institution of Naval Architects.

Dr. *Statement of Receipts and Payments for the Year* 1900. Cr.

| Receipts. | £ s. d. | £ s. d. |
|---|---|---|
| To Balance at Banker's, Dec. 31, 1899 ... ... ... ... | 437 14 10 | |
| „ Balance in Secretary's hands, Dec. 31, 1899 ... ... ... ... | 9 17 3 | |
| | | 447 12 1 |
| „ Annual Subscriptions ... ... ... | | 2,851 15 4 |
| „ Sale of Volumes ... ... ... ... | | 233 15 3 |
| „ Life Subscriptions... ... ... ... | | 126 0 0 |
| „ Admiralty Grant ... ... ... ... | | 250 0 0 |
| | | £3,909 2 8 |

| Payments. | £ s. d. | £ s. d. |
|---|---|---|
| By Rent ... ... ... ... | 220 0 0 | |
| „ Housekeeper and Cleaning ... | 37 14 0 | |
| | | 257 14 0 |
| „ Banker's Charges ... ... | 2 14 1 | |
| „ Stationery... ... ... ... | 37 10 6 | |
| „ Auditors' Fee ... ... ... | 5 6 6 | |
| | | 45 11 1 |
| „ Insurance... ... ... ... | 2 10 0 | |
| „ Petty Disbursements ... ... | 33 15 4½ | |
| „ Postages and Telegrams... ... | 44 13 9 | |
| | | 80 19 1½ |
| „ Printing Volume XLII., Papers, Circulars, &c., &c. ... ... | 761 12 6 | |
| „ Gold Medal and Premium ... | 21 16 0 | |
| „ On Account of Preparation of General Index ... ... | 50 0 0 | |
| | | 833 8 6 |
| „ Salaries ... ... ... ... | 1,275 19 11 | |
| „ Expenses of Spring Meetings ... | 64 4 6 | |
| „ Reporting ... ... ... | 50 0 0 | |
| | | 1,390 4 5 |
| „ Travelling ... ... ... | 42 19 1 | |
| „ Miscellaneous ... ... ... | 36 8 0 | |
| „ Despatch of Volumes ... ... | 75 19 2½ | |
| | | 155 6 3½ |
| „ Investments ... ... ... | 589 1 0 | |
| „ Furniture and Repairs of Office | 27 7 8 | |
| | | 616 8 8 |
| „ Balance at Banker's Dec. 31, 1900 | 457 9 0 | |
| „ Balance in Secretary's hands to meet Current Expenses, Dec. 31, 1900 ... ... ... | 72 1 7 | |
| | | 529 10 7 |
| | | £3,909 2 8 |

## LIBRARY FUND.

| Receipts. | £ s. d. |
|---|---|
| To Balance at Banker's, December 31, 1899 ... | 31 9 4 |
| Entrance Fees ... ... ... ... ... | 165 18 0 |
| Dividends on Investments ... ... ... | 212 15 0 |
| | £410 2 4 |

| Payments. | £ s. d. |
|---|---|
| By Library Expenses ... ... ... ... | 35 3 1 |
| „ Furniture and Repairs ... ... ... | 107 11 1 |
| „ Investments ... ... ... ... ... | 200 0 0 |
| „ Balance at Banker's, December 31, 1900 ... | 67 8 2 |
| | £410 2 4 |

H. MORGAN, Honorary Treasurer.
GEORGE HOLMES, Secretary.

We have examined the above-written entries with books and vouchers, and find them correct.

BALL, BAKER, DEED, CORNISH & Co.,
Chartered Accountants.

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**The following is a List of Donations to the Library:—**

"Annual Report of the Royal National Lifeboat Institution," 1900. *Presented by the Royal National Lifeboat Institution.*

"Bureau Veritas International Register of Shipping," 1901. *Presented by the Council of the Bureau Veritas International Register of Shipping.*

"British Corporation for the Survey and Registry of Shipping," 1897. *Presented by the British Corporation.*

"Bulletin de l'Association Technique Maritime, Congrès d'Architecture et de Construction Navales," Session 1900. *Presented by l'Association Technique Maritime.*

"Bulletin de la Société Scientifique et Industrielle de Marseille," 1900. *Presented by the Editor.*

"Bulletin de la Société d'Encouragement pour l'Industrie Nationale," for 1900. *Presented by the Société d'Encouragement pour l'Industrie Nationale.*

"Calendar of the University College of South Wales and Monmouth," 1900-01. *Presented by the Registrar of the University College of South Wales and Monmouth.*

"Festschrift zur 42 Hauptversammlung des Vereins Deutscher Ingenieure in Kiel," 1901.

"Germanischer Lloyd: International Register for the Year 1901. *Presented by the Committee of the Germanischer Lloyd.*

"Hamburg's Handel im Jahre 1900." *Presented by the Hamburg Chamber of Commerce.*

"Inertia Stress of Elastic Gears." *Presented by the Author*, J. H. MACALPINE, Esq.

"Jahrbuch der Schiffbautechnischen Gesellschaft." *Presented by the Schiffbautechnische Gesellschaft.*

"Jahresbericht der Handelskammer zu Hamburg, über die Jahre 1898, 1899, 1900." *Presented by the Hamburg Chamber of Commerce.*

"Journal of the Franklin Institute," for 1900. *Presented by the Franklin Institute.*

"Journal of the Iron and Steel Institute," for 1900. *Presented by the Iron and Steel Institute.*

"Journal of the Royal United Service Institution," for 1900. *Presented by the Royal United Service Institution.*

"Journal of the Society of Arts," for 1900. *Presented by the Society of Arts.*

"Journal of the United States Artillery." *Presented by the Editor of the Journal.*

"Journal of the Western Society of Engineers," 1900. *Presented by the Western Society of Engineers.*

"Katalog der Commerz-Bibliothek in Hamburg, 1864 to 1900." *Presented by the Hamburg Chamber of Commerce..*

"La Navigation Internationale et ses Intérêts," 3ᵉ Section. *Presented by the Author*, LINDON BATES, Esq. *Pamphlet.*

"La Tour de Trois Cents Mètres." *Presented by* Mons. E. EIFFEL.

"Lloyd's Register of British and Foreign Shipping," 1901-02. *Presented by the Committee of Lloyd's Register.*

"Lloyd's Register of Yachts," for 1901. *Presented by the Committee of Lloyd's Register.*

"Mathematical Drawing Instruments." *Presented by the Author*, W. F. STANLEY, Esq.

"Mémoires et Compte-Rendu des Travaux de la Société des Ingénieurs Civils de France," 1900. *Presented by the Société des Ingénieurs Civils de France.*

"Minutes of the Proceedings of the Institution of Civil Engineers," Vols. CXL., CXLI., CXLII., and CXLIII. *Presented by the Institution of Civil Engineers.*

Newspapers and Magazines:—

"Annalen für Gewerbe und Bauwesen," for 1900. *Presented by the Proprietor.*

"Arms and Explosives," for 1900. *Presented by the Proprietors.*

"Army and Navy Gazette," for 1900. *Presented by the Proprietors.*

"Automotor and Horseless Vehicle Journal," 1900. *Presented by the Proprietors.*

"Cassier's Magazine," for 1900. *Presented by the Proprietors.*

"Commerce," 1900. *Presented by the Proprietors.*

"Engineer," for 1900. *Presented by the Proprietors.*

"Engineering," for 1900. *Presented by the Proprietors.*

"Engineering Magazine," 1900. *Presented by the Proprietors.*

"Engineering Times," Vol. III. *Presented by the Proprietors.*

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Newspapers and Magazines—*continued.*

"Engineers' Gazette," 1900. *Presented by the Proprietors.*

"English Mechanic," for 1900. *Presented by the Proprietors.*

"Field," for 1900. *Presented by the Proprietors.*

"Fielden's Magazine," 1900. *Presented by the Proprietors.*

"Invention," 1900. *Presented by the Proprietors.*

"Iron and Coal Trades Review," for 1900. *Presented by the Proprietors.*

"Journal of the Imperial Institute," for 1900. *Presented by the Council of the Imperial Institute.*

"Le Moniteur de la Flotte," for 1900. *Presented by the Proprietors.*

"Machinery Market," for 1900. *Presented by the Proprietors.*

"Marine Engineer," for 1900. *Presented by the Proprietors.*

"Marine Engineering," 1900. *Presented by the Proprietors.*

"Marine Review," for 1900. *Presented by the Proprietors.*

"Mariner," 1900. *Presented by the Proprietors.*

"Mechanical Engineer," 1900. *Presented by the Proprietors.*

"Practical Engineer," for 1900. *Presented by the Proprietors.*

"Railway Engineer," 1900. *Presented by the Proprietors.*

"Saturday Review," for 1900. *Presented by the Proprietors.*

"Science Abstracts," from January, 1900. *Presented by the Proprietors.*

"Shipping World," for 1900. *Presented by the Proprietors.*

'Steamship," for 1900. *Presented by the Proprietors.*

"Thames Ironworks Quarterly Gazette." *Presented by the Proprietors.*

"Zeitschrift des Vereins Deutscher Ingenieure," 1900. *Presented by the Proprietors.*

"Photographs of Members of the Institution." *Presented by the Photographers*, Messrs. MAULL & FOX, Piccadilly.

"Proceedings of the Engineering Association of New South Wales," Vol. X., 1895.

"Proceedings of the Institution of Mechanical Engineers," for 1900. *Presented by the Institution of Mechanical Engineers.*

"Proceedings of the London Mathematical Society," 1900, and Index, Vols. I.—XXX. *Presented by the London Mathematical Society.*

"Proceedings of the Royal Society of London," Vols. LXVI. and LXVII., 1900.

"Proceedings of the Royal Society of New South Wales," Vol. XXXIII. 1899. *Presented by the Royal Society of New South Wales.*

"Proceedings of the United States Naval Institute," for 1900. *Presented by the U.S. Naval Institute.*

"Report of His Majesty's Commissioners for the Paris International Exhibition, 1900." Vols. I., II. *Presented by the Commissioners.*

"Report upon the Seventh International Congress of Navigation held at Brussels." *Presented by the State Department, Washington, U.S.A.*

"Report of Tests of Metals and other Materials made at Watertown Arsenal," for the year ending June 30, 1899. *Presented by the Chief of Ordnance, U.S.A.*

"Revista Marittima." *Presented by the Ministero della Marina.*

"Technology Quarterly and Proceedings of the Society of Arts," 1900. *Presented by the Massachusetts Institute of Technology.*

"The Naval Annual," 1901. By the Right Hon. Lord BRASSEY, K.C.B., and others. *Presented by the Right Hon.* Lord BRASSEY, K.C.B., D.C.L.

"The Norwegian North Polar Expedition, 1893-1896. Scientific Results. The Fram I." *Presented by the Author*, COLIN ARCHER, Esq. *Pamphlet.*

"The Scientific Proceedings of the Royal Society of Dublin," for 1900. *Presented by the Royal Society of Dublin.*

"The Studies and Preliminary Documents Relating to the Construction and Operation of a Commercial Port in the City of Rosario." *Presented by* Mons. Duilout, *Argentine Inspector.*

"Theorie des Schlick'schen Massen Ausgleichs bei mehrkurbeligen Dampfmaschinen," von Professor Dr. HERMANN SCHUBERT, Hamburg. *Presented by* Herr OTTO SCHLICK.

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"Transactions of the American Institute of Mining Engineers," Vol. XXIX. *Presented by the American Institute of Mining Engineers.*

"Transactions of the American Society of Mechanical Engineers," Vol. XXI., 1900. *Presented by the Council of the American Society of Mechanical Engineers.*

"Transactions of the Hull and District Institution of Engineers and Naval Architects." *Presented by the Hull and District Institution of Engineers and Naval Architects.*

"Transactions of the Institution of Engineers and Shipbuilders in Scotland," Vol. XLIII., 1899-1900. *Presented by the Institution of Engineers and Shipbuilders in Scotland.*

"Transactions of the Institution of Junior Engineers," Vol. X., 1900. *Presented by the Institution of Junior Engineers.*

"Transactions of the Institute of Marine Engineers," for 1899-1900. *Presented by the Institute of Marine Engineers.*

"Transactions of the Liverpool Engineering Society," for 1900. *Presented by the Liverpool Engineering Society.*

"Transactions of the North-East Coast Institution of Engineers and Shipbuilders," Vol. XVI., 1899-1900. *Presented by the North-East Coast Institution.*

"Transactions of the North of England Institute of Mining and Mechanical Engineers," for 1900. *Presented by the North of England Institute of Mining and Mechanical Engineers.*

"Transactions of the Society of Engineers," for 1900. *Presented by the Society of Engineers.*

"Transactions of the Society of Naval Architects and Marine Engineers," Vol. VIII., 1900. *Presented by the Society of Naval Architects and Marine Engineers.*

"Travaux Scientifiques Exécutés à la Tour de Trois Cents Mètres de 1889 à 1900." *Presented by* Mons. G. EIFFEL.

The following gentlemen, having been duly recommended by the Council, were unanimously elected members of this Institution:—Mr. William Ames, Chief Engineer, New South Wales Naval Force, New South Wales; Mr. Thomas Andrews, Assistant Manager to Messrs. Harland & Wolff, Ltd., Belfast; Mr. Frederick W. Baker, Chief Draughtsman New York Shipbuilding Co., New York; Mr. T. C. Billetop, Manager to Messrs. Henry Watson & Sons, Newcastle-on-Tyne; Mr. George H. Ball, First Class Assistant Constructor to the Admiralty, Whitehall; Mr. Charles J. Blackburn, Superintendent Engineer to the Isle of Man Steam Packet Co., Douglas, Isle of Man; Mr. Charles M. Burls, Partner in the firm of Messrs. E. J. Caiger & Co., Consulting Engineers and Surveyors, Billiter Buildings, E.C.; Mr. Harold E. J. Camps, Consulting Engineer and Naval Architect, Fenchurch Street; Mr. James L. Clark, Consulting Engineer and Marine Surveyor, Glasgow; Mr. Duncan Colquhoun, Board of Trade, Liverpool; Señor Enrique Garcia de Angulo, General of Naval Architecture and Director of Naval Construction, Spanish Ministry of Marine, Madrid; Mr. Charles S. Douglas, Assistant to the Professor of Naval Architecture in Glasgow University; Mr. William B. Fulton, Chief Draughtsman, Assistant Manager Royal Shipbuilding and Engineering Co., Flushing; Mr. Charles H. Gilbert, Engineer-in-Chief, Chilian Navy, Valparaiso; Mr. William Gray, Chief Draughtsman with Messrs. Denny Bros., Dumbarton; Mr. T. M. Greenip, Marine Superintendent for Manchester Liners, Ltd., Manchester; Mr. E. Hall-Brown, Senior Partner in the firm of Messrs. Hall-Brown, Buttery & Co., Govan, Glasgow; Mr. A. John Helyer, Manager to Messrs. Bellino, Fenderich & Co., Odessa, Russia; Mr. Archibald C. Holms, Surveyor to Lloyd's Register, Sunderland; Mr. James R. Jack, Assistant Chief Draughtsman to Messrs. Denny Bros, Dumbarton; Mr. Arthur Jackson, Assistant to the Superintendent Engineer and Constructor of Shipping, War Department, Pall Mall; Mr. James Laing, Superintendent and Consulting Engineer, Aberdeen; Herr Gustav Lendecke, Director of the Engineering Department, Stabilmento Technico, Triestino, Austria; Mr. John Lowe, Chief of the Designing Department of Messrs. Caird & Co., Greenock; Mr. John Lyall, Superintendent to Messrs. Cayzer, Irvine & Co., Glasgow; Mr. J. H. Macalpine,

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Kilmalcolm, Scotland; Mr. George M. Miller, Managing Director, Messrs. Napier & Miller, Ltd., Shipbuilders and Engineers, Yoker, Glasgow; Mr. James Piper, Piper's Wharf, Greenwich; Mr. G. Polley, Manager to the Stabilmento Technico Triestino, Trieste, Austria; Mr. George Richard, Engineering Manager to the Smith Dock Co., Ltd., South Shields; Mr. James Schofield, Senior Partner in the firm of Messrs. Hagerup & Doughty, The Shipyard, Grimsby; Mr. Thomas S. Short, Managing Director to Messrs. Short Bros., Ltd.; Mr. Thomas Skinner, Board of Trade Surveyor, Barry Dock, Barry; Mr. James A. Smith, Manager of the Hooghly Dockyard, Calcutta; Mr. George H. Stafford, of Messrs. Laird Bros., Birkenhead; Mr. William F. Thumm, of Messrs. William Cramp & Sons, Philadelphia; Mr. Thomas L. Twaddell, Shipyard Manager to Messrs. Palmer's Shipbuilding and Iron Co., Ltd., Jarrow-on-Tyne; Mr. J. A. Ubsdell, Manager to the Chicago Shipbuilding Co., South Chicago, U.S.A.; Mr. Mario Vallino, Principal Designer to Messrs. G. Ansaldo & Co., Sampierdarena, Italy; Mr. Frank L. Warren, Consulting Engineer and Marine Surveyor, Fenchurch-street, E.C.

The following gentlemen were elected Associate Members:—Dr. Gustav Bauer, Engineer, the Vulcan Works, Stettin; Mr. William D. Barber, Chief Draughtsman Messrs. Forrest & Son, Ltd., Wyvenhoe, Essex; Mr. James Bloomfield, Manager to the Boston Deep Sea Fishing and Ice Co., Ltd., Boston, Lincolnshire; Mr. Arthur E. Butterfield, Assistant to Mr. Martin Samuelson, Hull; Mr. Frederick E. W. Coller, Assistant to the Yard Manager of Messrs. Sir W. G. Armstrong, Whitworth & Co., Ltd., Elswick; Mr. George C. Cook, of Howaldtswerke, Kiel, Germany; Mr. Charles I. Davidson, Surveyor to Lloyd's Register of Shipping, Liverpool; Mr. Herbert W. Fairbrass, Naval Architect and Yacht Designer, Haymarket; Mr. William G. Gibbons, Manager of the Goole Branch of Earle's Shipbuilding and Engineering Co., Hull; Mr. Ellison Halder, Manager to Messrs. Renwick and Wilton, Dartmouth; Mr. Charles F. Hitchins, Consulting Engineer, Victoria-street, S.W.; Mr. George Howson, Engineering Draughtsman, Fairfield Engineering Co., Ltd., Govan, Glasgow; Mr. Arthur Morris, of the Drawing Office, Fairfield Shipbuilding and Engineering Co., Govan, Glasgow; Mr. Ernest H. Rigg, Draughtsman to the British Corporation for the Survey and Registry of Shipping, Glasgow; Mr. George W. Rogers, Managing Director of the Irvine Shipbuilding Co., Irvine, Ayrshire; Mr. J. C. Garcia de Sola, in the service of Messrs. William Denny and Bros., Dumbarton; Mr. Edward Wilding, Junior Assistant Constructor, Admiralty Experimental Works, Haslar, Gosport.

The following gentlemen were elected Associates:—Mr. J. M. Adam; Mr. H. G. G. Ashton; Captain R. H. Bacon, R.N.; Mr. R. E. Barry; Mr. A. N. Bostelman; Mr. T. Gibson Bowles, M.P.; Mons. G. Canet; Captain A. C. Corry, R.N.; Mr. Thomas Free; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.; Mr. James Harrington; Mr. Christopher Head; Mr. E. S. Hopkinson; Mr. G. F. Johnstone; Mr. John T. Lunn; Mr. L. A. M. McGeoch; Mr. James W. Miller; Mr. George Milne; Mr. C. E. Nicholson; Mr. P. D. Ouless; Mr. G. N. Philip; Colonel F. B. Rognetta; Mr. G. O. Spence; Mr. G. P. Spooner; Mr. R. E. Stephens; Mr. Oscar S. Thompson.

Since the issue of Volume XLII. the Institution has sustained the loss by death of the following gentlemen:—The Right Hon. Lord Armstrong, C.B., D.C.L., F.R.S., Vice-President; Sir Raylton Dixon, Member of Council: Mr. R. R. Bevis; Mr. T. H. Bridges; Mr. N. P. Edwards; Mr. Charles Hall; Mr. John Henderson; Mr. Charles Langtry; Sir Andrew Maclean; Mr. Thomas Menzies; Mr. James Moir; Mr. George Rickard; Mr. Charles Sells; Mr. Thomas Todd, Members; Mr. Orde Barry, Associate Member; Admiral of the Fleet Sir J. E. Commerell, G.C.B., V.C.; Vice-

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Admiral Murray Aynsley, C.B.; Mr. J. A. Brand; Mr. John Colville, M.P.; Mr. F. Glenfield; Mr. William Houldsworth; Mr. James Huddart; Herr C. F. Laeisz; Mr. G. A. Laws; Captain G. H. C. MacArthur, R.N.; Mr. C. W. Parry; Mr. Arthur Price, R.N., Associates.

The SECRETARY next read the following list of names of the retiring Members of Council and the new names nominated by the Council to fill up the vacancies for the Vice-Presidents and for the ordinary and Associate Members of Council for the ensuing year. Names proposed for election of Vice-Presidents:—Professor J. H. Biles; Sir Raylton Dixon; Mr. John Scott, C.B.; Mr. Philip Watts, F.R.S. Retiring Members of Council:—Professor J. H. Biles; Mr. R. J. Butler; Mr. A. Denny; Mr. D. J. Dunlop; Mr. J. Macfarlane Gray; Mr. A. E. Seaton. New candidates nominated for Members of Council:—Mr. A. Gracie; Mr. A. Holt; Mr. Thomas Soper; Mr. H. F. Swan; Mr. H. H. West; Sir James Williamson. Retiring Associate Members of Council:—Mr. R. E. Froude; Vice-Admiral G. D. Morant. New candidates nominated for Associate Members of Council: —The Hon. T. A. Brassey; Rear-Admiral C. C. P. FitzGerald.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.) next put to the meeting the following list containing the names of the President, Vice-Presidents, and Treasurer for the ensuing year, which was unanimously adopted. President—The Right Hon. the Earl of Glasgow, G.C.M.G., LL.D. Past Presidents—The Right Hon. the Earl of Ravensworth; the Right Hon. Lord Brassey, K.C.B., D.C.L.; the Right Hon. the Earl of Hopetoun, P.C., K.T., G.C.V.O., G.C.M.G. Vice-Presidents—H.R.H. the Duke of Cornwall and York, K.G., K.T., K.P., G.C.V.O., G.C.M.G., Rear-Admiral; the Right Hon. the Earl of Northbrook, G.C.S.I.; the Right Hon. the Earl of Ravensworth; the Right Hon. Earl Spencer, K.G.; the Right Hon. Lord George Hamilton, M.P.; the Right Hon. Lord John Hay, G.C.B., Admiral of the Fleet; the Right Hon. Sir John Dalrymple-Hay, Bart., K.C.B., D.C.L., F.R.S., Admiral; Sir Nathaniel Barnaby, K.C.B.; Sir Frederick Bramwell, Bart., D.C.L., F.R.S.; Sir John Durston, K.C.B., R.N.; Sir John Glover; Sir Andrew Noble, K.C.B., F.R.S.; Sir Edward J. Reed, M.P., K.C.B., F.R.S.; Sir W. H. White, K.C.B., LL.D., Sc.D., F.R.S.; F. K. Barnes, Esq.; James Dunn, Esq.; Francis Elgar, Esq., LL.D., F.R.S.; John Inglis, Esq., LL.D.; Benjamin Martell, Esq.; Henry Morgan, Esq. (Treasurer); George W. Rendel, Esq.; J. I. Thornycroft, Esq., LL.D., F.R.S.; A. F. Yarrow, Esq.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): It is with some diffidence that I rise to say a few words to you, and they are, that the Members of your Institution who go to Glasgow will, I presume, on this occasion as upon others, when they visit different ports and different places in the world, combine a little pleasure with their business. They probably will have some excursions in the country. I only want to say that my house and property at Kelburne are situated where a most beautiful view of the Clyde is to be obtained, and it will give me the greatest pleasure if a day may be found when I can give a garden party where you will be able to enjoy that beautiful view. It is now my duty, and pleasure, to present to Professor Bryan the Gold Medal for the year 1900 for his Paper "On the Action of Bilge Keels."

Professor G. H. BRYAN, M.A., F.R.S.: My Lord, I can only say how deeply I appreciate the honour you have conferred upon me by awarding me this Gold Medal. I feel that it may well be the ambition of a Cambridge mathematician to earn such recognition at the hands of a body of practical men. I cannot express to you at all adequately how deeply I feel the value of the recognition which you have conferred upon me.

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The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.) then proceeded to deliver the following Opening Address:—In rising to address you, my first duty is to offer you my most sincere thanks for the honour you have done me in electing me to fill your Presidential chair. I can assure you from my heart that no efforts on my part will be wanting, in order that I may perform to your satisfaction the duties of the office. It would be difficult for anyone to succeed Lord Hopetoun, whose charm of manner, courtesy, tact, and ability, have enabled him to fill so successfully every position which he has hitherto held: I am sure I am only echoing the sentiments of all present in wishing him every success in the discharge of the arduous duties of the high office to which he has been called by his Sovereign.

Gentlemen, I am profoundly impressed by the importance of the Institution with which I am here associated. Largely dependent as we are on our seaborne commerce, and on our Navy, the Institution of Naval Architects must ever be one of the foremost among the great interests of the country, and its prosperity and that of the Empire may be said to be inseparable. I consider that it is a thing to be proud of, to have the honour of presiding over your deliberations.

Before turning to the business part of my address, I feel that it would be your wish that I should make some allusion to the great bereavement which the nation has sustained. By the death of our late most Gracious Sovereign, we have lost one of the greatest monarchs who ever sat on a throne: a pattern for all constitutional rulers; the longer she lived, the more she was beloved. Of the many virtues she possessed, perhaps the greatest was that of sympathy. She was ever ready, on all occasions, to show that the nation's joys and sorrows were hers. It will be long before we lose the sense of personal loss we all feel. Forget her we never shall.

We have also to deplore the loss of H.R.H. the Duke of Saxe-Coburg. He was a true friend of the Navy, an excellent officer, and ever had the interests of the Service at heart.

The Institution has also to mourn the loss of one of its oldest, best known, and most respected members, in the death of the late Lord Armstrong. Lord Armstrong's services in the fields of hydraulic engineering and of artillery will secure for him a high place in the annals of British engineers, and the great establishment which he reared on the banks of the Tyne will be an enduring monument of his genius and capacity.

While it is only becoming to take note of the losses the Institution has sustained by death, we must not omit to record our deep sense of regret at the prospective loss which we are about to suffer of the services of our much-esteemed Secretary. Mr. Holmes has served you in that capacity for twenty-two years. An engineer by profession, an excellent linguist, a man of exceptional ability, Mr. Holmes is also a most accomplished man, and, above all, an ideal Secretary. My experience of the value of his services has been brief, but it has been sufficient to give me some idea of the loss the Institution sustains by his acceptance of the important post of Chairman of the Board of Public Works of Ireland. I am sure I am expressing the feelings of all of you, when I say that, while we rejoice that His Majesty, through his advisers, has been pleased to appoint Mr. Holmes to an important position, we shall part from him with the utmost regret, and we wish him every success and happiness in his new sphere of activity.

It is customary for your President in his Annual Address to refer to the progress made during the year in the Royal Navy and the Mercantile Marine. At the commencement of a new century, it

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would be a fascinating task to trace out and put on record the vast changes which have taken place during a hundred years in the types, materials of construction, dimensions and means of propulsion, armament and equipment, in warships and mercantile vessels. Such a record would be too vast to form the subject of a Presidential Address: but, I venture to suggest that "A Century of Shipbuilding" would form a peculiarly appropriate title for a paper to be read this year at our forthcoming Summer Meeting on the Clyde, the river which has given birth to so many of the changes which have made the nineteenth century memorable in the annals of your profession, and I venture to hope that, amongst you may be found some historian versed in the technicalities of the craft, who will give the Institution a paper on this subject which will serve as a work of reference for future writers.

I must confine myself to a review of the history of the past year.

Turning, first, to the Mercantile Marine, it is interesting to note that the shipbuilding output of the United Kingdom in the last year of the century has surpassed all previous records. Of merchant vessels no less than 1,442,471 tons were launched, made up of 664 steamers, and only 28 sailing vessels. These figures surpass those of the previous year by 26,000 tons, and the average of the preceding decade by no less than 326,000 tons. The warships launched amounted to only 168,000 tons displacement, having been less, by a round 100,000 tons, than the output of the previous year. Only about half of this amount was for the British Navy, but this fact must be regarded as a mere accident, for many of H.M. warships were approaching the launching condition at the end of the year, and within the last few weeks no less than seven of the largest size have been put afloat. The figures, therefore, given above, do not in any sense represent the activity of the country in warship building during the past year.

It is interesting to note what becomes of the huge tonnage of mercantile shipping which is constructed every year in these islands. It appears that rather more than three-quarters of the whole is for home owners, and rather less than one quarter—to be exact, about 23 per cent.—goes to our Colonies and to foreign countries. Germany and Austria-Hungary were our largest foreign customers, having taken 103,000 and 80,000 tons respectively.

The most remarkable feature of modern mercantile shipbuilding is the continually increasing size of individual steamers. In 1892 there were launched in the United Kingdom 37 steamers, of over 4,000 tons gross, while last year 125 were built. In the first-named year there was only one steamer put afloat of over 7,000 tons, against 26 in the year 1900. If we go to the very largest type, it may be noted that there are now 13 steamers of 10,000 tons and upwards being constructed in these islands, against 3 in Germany, 1 in France, and 6 in the United States. Of these 13 vessels, 5 are stated to exceed 18,000 tons each.

I must here refer specially to one of these steamers; because, when completed, she will be the largest ship in the world. The *Celtic*, which will be launched by Messrs. Harland & Wolff, in Belfast, next week, exceeds even the *Oceanic* in tonnage. Her leading dimensions are:—Length over all, 698 ft. 8 in.; breadth, 75 ft. 4½ in.; depth, 49 ft.; gross tonnage, 20,900. It will be noted that, while the length is about 7 ft. less than that of the *Oceanic*, the breadth is 7 ft., and the gross tonnage 3,600 in excess. The horse-power is only one-half that of the latter vessel, being 14,000 against 28,000 I.H.P. This is accounted for by the fact that the *Celtic* is of the combined passenger and cargo type, whereas the *Oceanic* is a mail steamer.

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Outside of the United Kingdom the shipbuilding industry has been extremely active. In France, the operation of the Bounty Laws has continued to foster the production of many large sailing ships. In the United States there has been an enormous development of shipbuilding capacity; the ocean tonnage, in iron and steel, reported as being under construction has very nearly doubled, having risen from 105,000 tons at the end of 1899, to 198,000 at the end of last year. This is altogether exclusive of the shipbuilding industry on the Great Lakes, which, as is well known, has already attained great proportions, the iron and steel vessels constructed for these inland waters having, last year, reached the total of 143,000 tons. The Americans are making great preparations to carry on shipbuilding on a very large scale, and when the threatened bounty laws are passed it seems probable that we in this country shall once more have to face the competition in our most distinctive national industry, which was felt so severely in the last decades of wooden shipbuilding.

It is well worthy of remark that in this the first year of the New Century, the wooden shipbuilding industry continues to flourish in the United States. Last year no less than 107,500 tons gross were constructed of timber. Amongst them may be noted the *Eleanor A. Percy*, a six-masted schooner, built at Bath, in the State of Maine. She is probably the largest sea-going sailing vessel ever built. Her leading dimensions are as follows:—Length, 323 ft. 6 in.; breadth, 50 ft.; depth, 24 ft. 9 in.; gross tonnage, 3,402. The schooner *Pretoria*, launched last July at West Bay City, Michigan, is even longer than the *Eleanor A. Percy*. Her dimensions are: Length, 350 ft.; breadth, 45 ft. 6 in.; depth, 27 ft. She is intended to carry 5,000 tons of ore, or 175,000 bushels of wheat.

In Germany shipbuilding continues to flourish. Over 204,000 tons of merchant vessels were constructed there last year. We all know the enterprise and skill of our German friends, especially in building Transatlantic Liners of the largest class. The famous Hamburg-American steamship, the *Deutschland*, holds the Atlantic record, while the North German Lloyd steamer, *Kaiser Wilhelm der Grosse*, follows closely in the second place.

The statement recently made to Parliament by the First Lord of the Admiralty is full of interest. I must congratulate Lord Selborne, and also his predecessor, upon the fact of "better progress having been made than in recent years with the ships under construction for the Royal Navy." For the first time for some years "the aggregate expenditure on new construction will closely approach the provision made in the estimates." Last year, from this chair, your then President, Lord Hopetoun, felt constrained to comment on the serious national danger that had arisen through the non-delivery of machinery and armour, and suggested that the Government should either encourage private firms to lay down plant, or else should itself undertake the manufacture. It is very satisfactory to know that the private firms have largely increased their appliances, and that the number of manufacturing firms has increased to five. It now seems probable that delays will not again arise in consequence of deficient producing power.

Lord Hopetown also alluded last year to the necessity of auxiliaries, such as steam colliers, repairing, distilling, and hospital ships being provided for the use of the fleet. All must recognise the importance of these additions, which increase the mobility of the Navy and its readiness for war. It is satisfactory to note that some progress has been made towards providing these vessels. Three colliers are now in use with the fleet; a ship for repairing and distilling purposes has been



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purchased, and is being fitted up; another distilling ship and also a depôt ship are to be built; while the *Maine*, which rendered such signal service in South Africa, will, it is hoped, be secured as a hospital ship in the Mediterranean. The whole question of fleet auxiliaries is now under the careful consideration of the Board, and it is earnestly to be hoped that, next year, we may hear that substantial progress has been made in their provision.

A very interesting feature in the statement is the announcement that, for the first time in our history, submarine boats have been ordered for experimental purposes. Experts differ widely in their views as to the effect that these boats may have upon the future of naval warfare; but no one can doubt that the Admiralty was wise in resolving to experiment with them. It must be regarded as a fortunate coincidence that we are to have at these meetings an important paper on the subject of submarine boats and their manipulation, from Captain Hovgaard, of the Royal Danish Navy, an officer who has already made many interesting communications to the Institution, and whose presence among us is always welcomed.

One of the most interesting events of the year has been the inquiry which has been entered into by the Boiler Committee, a body composed very largely of distinguished members of this Institution. Into the knotty technical questions concerning the merits and demerits of different types of boilers for warships I do not propose to enter; but, I may be allowed to express the hope that the labours of the Committee will result in great benefit to the Navy, and that the Admiralty will permit the vast masses of information, which will doubtless be accumulated, to be rendered available for the profession.

The Admiralty programme of shipbuilding is always looked forward to with great interest by the members of this Institution. For the coming financial year, the sum of a little over nine millions sterling has been asked for new construction. The greater part of this amount is to be devoted to pushing on the ships now being built, as far as possible, towards completion. No one can question the wisdom of this policy. The vote demanded is the largest on record, but is not, in my opinion, greater than the circumstances demand. Some disappointment has been expressed at the smallness of the sum (£537,850) to be devoted towards commencing an absolutely new programme: but, if it has been found impossible to realise simultaneously both the objects in view, viz., to make the utmost possible progress with the work actually in hand, and at the same time to make a substantial beginning with new construction, then, I think that the wisdom of the Admiralty, in distributing the money as they propose to do, cannot be gainsaid. A good ship in the water is worth two on the stocks and any number on paper; a fact which is not, invariably, given due weight to by some of those who enlighten the country on naval matters. Whether, or no, it is impossible to realise both these objects simultaneously, I am not in a position to judge; but, no doubt, light will be thrown on this subject during the progress of the debates in Parliament on the Navy Estimates.

The new shipbuilding programme comprises three battleships, six armoured cruisers, two third-class cruisers, and twenty-two smaller vessels. The types of the new battleships and cruisers are, naturally, of great interest to us as an Institution; but, as the designs are not yet completed, I am unable to give you any information on this subject.

I am very glad to know that during my first year of office the Institution proposes to hold a Summer Meeting in my own country and district. I can assure you of a most cordial welcome to Glasgow. The Institution of Naval Architects is held in high honour on the Clyde. Its ship-

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builders, like all others, have derived great advantage from your labours, and you may be perfectly assured of their sympathy and support.

While on this subject I may mention that many of our members and friends from Germany, propose to visit the Great Exhibition which is to be held in Glasgow, and they will take advantage of the opportunity to assist us at our Summer Meeting. We can never forget the welcome which our German colleagues and their illustrious Monarch extended to us when we visited their country in 1896. Recent events have given that Monarch a very large place in the heart of the British people, and I need hardly say that we shall be delighted to welcome to our Meetings those of his people who take an interest in our pursuits. I hope also that our members in France, whose great services to us are mentioned in the Annual Report which has just been read to you, will not be so much spoilt by the glories of their own recent Great Exhibition as to make them unwilling to visit our lesser show in Glasgow this summer. I can assure them that, if they come, they will find much that is worth inspecting, especially in our own particular sphere of interests. They will also find that we are only too anxious to return the goodwill which they have so often shown to us. They will find that the old sympathy which existed between their country and Scotland is still in existence, and I can promise them a cordial welcome from my countrymen. I trust, also, that our members from the other European States and from America will not forget us on this occasion. We derive great benefit from their assistance. They continually give us papers of the highest merit; and I can assure them that we shall be very glad to see them, either at our own Summer Meeting, or at the International Engineering Congress, Glasgow (1901), the section of which, devoted to the interests of our profession, will be managed by ourselves.

Gentlemen, thanking you again for the honour you have done me, I conclude by moving the adoption of the Report.

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RESULT OF THE BALLOT FOR THE ELECTION OF VICE-PRESIDENT, MEMBERS, AND ASSOCIATE MEMBERS OF COUNCIL.

Before the reading of the first paper on Thursday morning, the Scrutineers appointed to examine the ballot papers for the election of the new Council, presented the following report :—

*Institution of Naval Architects*, 5, *Adelphi Terrace, London, W.C.*,
*March* 28, 1901.

MY LORD,—Having counted the votes for the election of a Vice-President, Members of Council, and Associate Members of Council, we find that the following have been elected :—

As Vice-President—

Mr. Philip Watts, F.R.S.

As Members of Council—

| | |
|---|---|
| Mr. Archibald Denny. | Mr. R. J. Butler. |
| Mr. J. Macfarlane Gray. | Mr. D. J. Dunlop. |
| Professor J. H. Biles. | Sir James Williamson. |

As Associate Members of Council—

Mr. R. E. Froude, F.R.S. The Hon. T. A. Brassey.

In consequence of the election of Mr. Watts as Vice-President, there is an additional vacancy on the Council, to which Mr. A. Gracie is elected.

We are, my Lord,

Your obedient Servants,

(Signed) H. J. CORNISH, Member of Council,
STANLEY TATHAM, Member,
(Scrutineers).

*To the Right Hon. the* EARL OF GLASGOW, G.C.M.G., LL.D., *President, Institution of Naval Architects.*

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# TEN YEARS' NAVAL CONSTRUCTION IN THE UNITED STATES.

By Professor J. H. Biles, Member of Council.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 27, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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In 1891, I had the honour to present to this Institution a paper, entitled, "Some Recent Warship Designs for the American Navy." This paper gave, in outline, a description of the modern ships existing in the Navy, the first one having been made in 1883. It also gave, in outline, a comparison of some of these ships with ships which were then building, or had been recently completed, for other navies. The paper was, however, chiefly devoted to a description of the ships that were projected in the end of 1889, and that were then all in their initial stages of construction. I ventured to say that the American ship designers and builders had been very capable in producing vessels quite equal to their promises; though there were at the time people who thought that, in some respects, the Americans were rather sanguine in their anticipations. I have thought it may interest the Members of this Institution to bring together facts, as far as I have been able to obtain them, to show to what extent the promises made when my former paper was read have been realised, and, in addition, to give as much information as I can of the intentions in designing of the American Naval Constructors. If it can be shown that their promises have been fulfilled, it is a fair assumption that what they now propose will be fully realised.

Some of the information given has been obtained from Admiral Hichborne's papers read before the American Society of Naval Architects, but most of it has been obtained from official documents published by the Navy Department of the United States, which corresponds to our Admiralty. All interested in warships and their construction owe a debt of gratitude to this Department of the Government for the generous manner in which it gives information to the people. In this country our professional newspapers to some extent do the work that is done by the Navy Department of the United States. Probably a part of the data contained in this paper to-day has been already published in our professional papers, but it may not be out of place to bring some of this information together in this Institution, to afford an opportunity of discussing interesting points in the American Naval Constructors' designs, as well as to give



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information to many who have not the time nor the opportunity of following connectedly and closely these developments of naval construction in the United States.

In order to understand why so much information is made public by the United States authorities, it is desirable to remember that no ship of war is built in the United States without a special Act of Congress being passed authorising its construction. This Act may or may not, but generally does, contain specific details as to the kind of ship and its leading characteristics. Frequently its cost, displacement, speed, armour, armament, and coal-carrying are defined in the Act, and are arrived at after full discussion, both in the Committee and in the House. In this way all who are interested have an opportunity of hearing what is said, and, if they wish, of doing what they can to influence the formation of the Act which fixes the design of the vessel. After the Act is passed, the Naval Department prepares the designs for the ship in conformity with the Act, and the various details are fully discussed between the different Bureaux. Plans are then issued to builders to tender upon, and alternative designs are invited from the builders.

If, in submitting their tenders, the builders submit alternative designs, these designs are considered in relation to the Act of Congress and to the designs prepared in the Department, and, inasmuch as the builder has to be responsible for the speed and draught of the vessel, there is no insurmountable difficulty in his getting his own alternative design adopted, if it is as good as that of the Department. It is evident that this process will produce a design upon which the greatest possible amount of discussion has been expended, and whether the method of design by discussion is a successful one, may be judged, to some extent, by the productions of the United States Navy Department.

Before describing the new ships, it will be desirable to refer to those which were projected at the date of my last paper.

In 1891 the ships which were projected were referred to in Table B (page 45, Vol. XXXII.) of my paper. I annex this Table to this paper, with the figures of the ships as actually constructed; but, instead of the numbers by which they were known, I have given the names.

It will be seen from this Table that there was one class of battleship having three examples; five types of cruisers, varying from 8,000 to 2,000 tons displacement, and a harbour defence ram. There were some other vessels referred to, but not included in the Table. They have been added to the new Table, and are the following:—A small type of battleship called *Texas; Maine* (whose unfortunate fate of later years is fresh in the memory of the whole world), a low freeboard monitor, *Monterey; Cushing*, a small torpedo boat. In Tables that follow, C, D, E, and F, are shown the corresponding lists of vessels projected and built since 1891; the names of ships of each class are

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given. By means of these Tables a comparison can be made of the number of classes and the numbers of ships in each class, so that the relative magnitude of the United States Navy, as projected, or completed, to-day, will be seen compared with what it was ten years ago.

The numbers, types, and total displacements are summarised in Tables at the end of the paper. Plates illustrating the vessels referred to in this paper are annexed. From these figures it may be deduced that the total tonnage projected, or built, in 1891 was 165,646 tons, costing $46,588,000 (£9,700,000); while the total tonnage to-day, projected, or built, is 353,400 tons, costing $96,613,000 (£20,127,000). These figures are very impressive and interesting to statesmen or politicians. While the Members of this Institution cannot fail to be also very much interested in the progress of the power of the American Navy, it is not within the intended scope of this paper to deal with that side of the question, but rather, after noting it, to consider the United States Navy from the Naval Constructors' point of view. It will be seen that the types group themselves in a manner not very unlike that of our own Navy. There has been no attempt to develop more than one class of battleship, though, naturally, with the lapse of time and the evolution of ideas the type changes somewhat; but it is evident that the American Naval Constructors have never contemplated building two classes of battleships at the same time. They always have intended to build battleships superior to those already built, and never have made any provision for another type of battleship that could be in any sense, from their point of view, called a second-class battleship. This may be seen from the following list of displacements and types of battleships built since 1889.

| Name. | Displacement. | Date of Authorisation. |
|---|---|---|
| *Indiana* ... ... | 10,200 | 1890 |
| *Iowa* ... ... | 11,340 | 1892 |
| *Kentucky* ... | 11,540 | 1895 |
| *Alabama* ... | 11,565 | 1896 |
| *Maine* (new) ... | 12,400 | 1898 |
| *Georgia* (sheathed) | 15,320 | 1899 |
| *Virginia* (unsheathed) | 14,950 | 1900 |

The last two are practically the same design, but the 1899 ships were sheathed, thus making their displacement slightly greater than the 1900 ships.

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The same may be said with reference to speed, but in no case has a later ship been designed to have a less speed than an earlier one, the intended speed of the latest ship being 19 knots, as against 15 knots in 1890.

With reference to armament, the same intention to have more powerful guns and more of them than in other battleships exists in the minds and in the designs of the American Naval Constructors. It is true that, comparing the *Indiana*, the 1891, class with the *Virginia* and *New Jersey* classes, the 1899, there is a reduction in the size of the big guns from 13 in. to 12 in.; but the guns of all sizes are more powerful, and, naturally, are much improved, as have been those of all other nations during the same period; but the number of 6 in. guns has increased from four to fourteen.

Turning to protection, improvements in the character of armour have enabled the United States Naval Constructors, like all others, to reduce thicknesses; but they have not made (until the last designs) the decided change in the arrangement of protection which has been adopted in our Navy. We have adopted much thinner side armour, and have associated with it the dome-shape form of protective deck which used to be peculiar to cruisers. But they still adhere to the dispositions, which existed ten years ago, of having the thickest armour at the water-line in a belt extending from 5 ft. below to about 3 ft. above load water-line, but with a much thinner belt between this point and the main deck. This arrangement has been, however, somewhat modified in the relative thicknesses, as, in the case of the *Indiana*, the armour was 18 in. at the water-line and 5 in. above; but in the later vessels of this type, the *Maine* class, there is 12 in. at the water-line and 6 in. above. In the latest, the *Virginia* class, the armour at the water-line is 11 in., and 6 in. above, but there is a dome-shape protective deck 3 in. on the slope and 1½ in. on the flat.

In the disposition of armour for gun protection, there have been considerable changes. In the 1891 ships, the *Indiana* class, the 13 in. guns were protected with turrets, having sloping sides of armour 17 in. thick, the barbette, or base of the turret, being protected also by 17 in. armour, while the 8 in. guns (of which there were eight in number in four pairs) were in turrets having 8 in. armour on the front, and 6 in. on the rear, with their bases protected by 10 in. armour part of the way down, and by much thinner armour in way of their ammunition tubes. The four 6 in. guns were in casemates on the main deck, having 5 in. armoured fronts and 2 in. rears. In the next type of ship, the *Iowa*, practically the same arrangement of guns and armour is carried out, except that the forward 12 in. guns are much higher, but the turret and barbette armour was reduced to 15 in. In the *Kentucky* and *Kearsage*, the next type, we see the abandonment of four of the 8 in. guns, and the substitution of the four 5 in. guns by fourteen 5 in. guns. The 13 in. guns had turrets the same as the *Iowa*, with 15 in. front plates and 12 in. at the rear,

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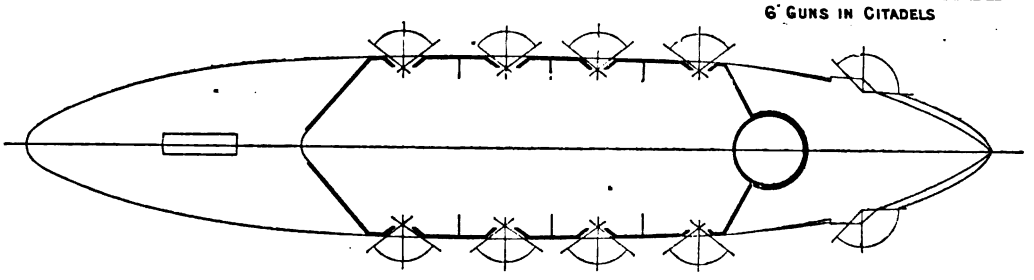
the barbettes being 15 in. thick at the front and 15 in. at the rear. But the remarkable feature of these designs was the novel arrangement of the double turrets, in which the 8 in. guns were superposed upon the turrets of the 13 in. guns, the whole being rotated by the same mechanism. This startling innovation caused a great deal of discussion in the United States, and has been generally approved of by the Ordnance Officers, but has been strongly opposed by the Naval Constructors. Further reference will be made to this question in describing some of the later ships.

The fourteen 5 in. guns were mounted on the broadside on the main deck within an armoured superstructure, or box battery. This also is an unusual arrangement of guns, though it is only a reversion to the ordinary arrangement of the earlier ironclads.

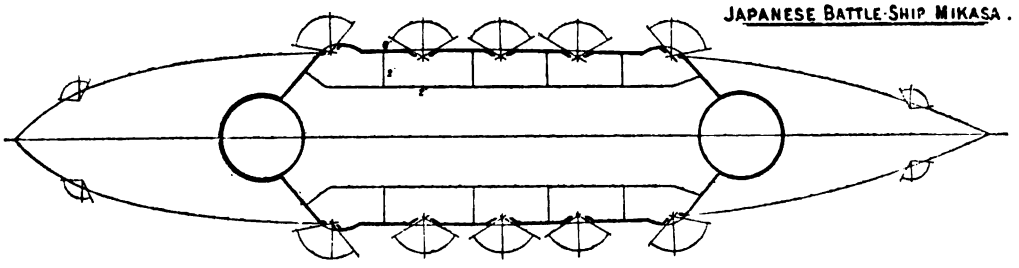
PLAN OF 6" BATTERY.

DESIGN FOR U.S. BATTLE-SHIP.
MAINE, MISSOURI & OHIO.
6" GUNS IN CITADELS



It is heavier than the arrangement of casemate adopted in our Navy, probably to the extent of 200 tons on a battleship of this kind, and, even with this extra weight, it reduces the arc of training of the guns. This arrangement has the advantage of bringing

JAPANESE BATTLE-SHIP MIKASA.



the guns' crews more under the control of the gunnery officers of the ship, and thus increases the tactical advantage of the battery; but against this must be set the advantage in the casemates of isolation of the guns' crews, which is no doubt great in case of the

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disablement of any one gun's crew. It is to be remarked that, while the protection, or freedom from penetration, is much better in the box battery than in the casemate arrangement, as the guns are protected in rear as well as in front, the damage is much greater in the event of penetration of the side by shell in the former than in the latter case, as probably in the former the whole of the battery's crew would be placed *hors de combat.* The arrangement of the *Mikasa,* which is building at Barrow to Mr. Dunn's design, for the Japanese Government, practically combines the advantages of the protection of both systems by the method shown in the figure on the previous page, in which each gun's crew is separated from the other; the protection in rear is better than in front, and must necessarily be much better than in the casemate system. The tactical advantage of control of all guns' crews at once is in this case surrendered.

In the next type of ship which was built, the *Alabama* class, the 8 in. guns were completely done away with, and the fourteen 5 in. guns were replaced by fourteen 6 in. guns, otherwise the arrangement was practically the same. This battery is of the same power as in our ships, but the 6 in. guns on the main deck are all in a box battery, as in the *Kentucky.* Four are in an open battery on the upper deck, where they have protection only on their fronts, and not in the rear; and two others are forward on the main deck, also having front protection only. The normal coal supply of the *Alabama* is 800 tons, as against 400 tons in the *Kentucky.* This change, together with the changing of 5 in. guns to 6 in. guns, has necessitated the doing away with the four 8 in. guns, so that the armament of the *Alabama* seems, in the opinion of the Naval authorities, to represent a stage at which they had practically agreed with the British Naval authorities in their ideas as to the best arrangement of armament, except in that part in which they prefer to adhere to the box battery rather than to have casemates.

In the next type of ship, the *Maine, Missouri, Ohio,* we have almost a repetition of the *Alabama* in the battery arrangement. The important change in these two ships, however, is the amount of speed and coal supply, the former being increased from 16 knots to 18 knots, while the latter is increased from 800 tons to 1,000 tons, the normal displacements being increased from 11,500 tons to 12,500 tons, in order to obtain these results. It is to be noted, however, that, practically, all the earlier battleships attained 17 knots on trial, and that it is probable that the *Maine* class will not be two knots in excess of the earlier classes. The last two classes, the *Georgia* and *Virginia,* have their largest guns reduced to 12 in., but their armament is very much strengthened by a battery of 8 in. guns eight in number.

The latest information on this subject seems to be that all the ships of both these classes are to be fitted with 8 in. superimposed on the 12 in. guns, and also a pair of turrets, one on each broadside, each having two 8 in. guns. There seems to have been

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a great conflict of opinion as to whether the arrangement of the *Indiana* or the arrangement of superimposed turrets should be adopted, but the latter arrangement seems at present to be most favoured. The thickness of armour on the turret in this latter class is reduced to 12 in. on the front and 11 in. to 9 in. on the rear in the turret for the 12 in. gun, while the barbette has 10 in. to 6 in. On the 8 in. turret the thickness is 8 in. to 6½ in. forward and 6½ in. to 6 in. aft. The 6 in. guns are twelve in number, and are arranged in a box battery on the main deck. These vessels are to have a displacement of 14,900 tons, or 15,300 tons, if sheathed; and are to have a speed of 19 knots. They have a coal supply of 900 tons, and a bunker capacity of 1,900 tons. One cannot fail to be struck with the enormous battery of these ships, and with the great protection which it has. It may, also, be noted that their draught is comparatively small, being only 24 ft. They are to have Babcock & Wilcox water-tube boilers. It is true, therefore, that the tendency has been to increase the size of battleships, and to have no second-class battleships, but rather to allow the older first class to become, by the lapse of time, in some sense second class. There has, however, been created for harbour defence purposes four 3,000 ton monitors. These are fully described in the Tables given, and in the illustrations.

These vessels are actually being built somewhat larger than the diagrams show. They have two 12 in. guns in one turret and four 4 in. guns on a superstructure. The side armour is 11 in. at the top, tapering to 5 in. at the lower edge, which is 5 ft. below the water. The upper edge is 3 ft. above the water. The armour of the turrets is 10 in. thick, and, unlike the earlier monitors, it has a barbette raising the turret on the upper deck of the ship. This raises the centre of the 12 in. gun, so that it is 10 ft. above the water-line, and 7 ft. above the deck. The openings round the funnels and ventilators are protected by 6 in. armour. There is a superstructure built up amidships, extending to within 8 ft. of the side of the ship, and which is over two decks in height. The lower deck carries the 4 in. guns, and the upper some smaller guns, and also affords additional accommodation of a very much better character than is usually the case in monitors for those who live on board. These vessels are only intended to steam about 12 knots, and have a small coal supply of 200 tons, the mean draught being only about 12 ft. 6 in. They have the defect common to all monitors—that the opening of almost any one compartment to the sea would, in all probability, cause them to sink. It seems to be quite unnecessary, in this class of ship, to put in anything like the elaborate subdivision that exists in them.

The results of the trials of these vessels that have been completed are given, and the ordinary particulars with reference to the new vessels, as far as they are known, are also set forth. Up till the time of the *Maine* class, 1898, there was no attempt made to adopt water-tube boilers in the battleship, but in the *Maine* twenty-four Niclausse boilers are to be fitted, and in the *Missouri* and *Ohio* there are twelve

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Thornycroft boilers in each. The *Georgia* class and the *Virginia* class are to have water-tube boilers. The engines are to attain 19,000 I.H.P. They have a 48 in. stroke, and run at about 110 to 120 revolutions, but in the *Maine* class the stroke has been reduced to 42 in. In the four monitors there are water-tube boilers, each ship having a different type, Thornycroft, Niclausse, Mosher, and Babcock & Wilcox. The results of these four vessels will be watched with a great deal of interest.

Turning to the cruisers, the *Brooklyn* was built in 1892, practically a sister ship to the *New York*. Nothing more was done in large cruisers until in 1899, when nine were projected, and in 1900 a further six. Dealing with them in the order of their displacement: the *California* class, three of which were ordered in 1899, are vessels very similar in dimensions to our *Drake* class, being about 500 ft. by 70 ft., with a draught of 24 ft. They are intended to steam 22 knots, and are to have 23,000 H.P. These vessels are armed with four 8 in. and fourteen 6 in. guns. They have a complete belt about 7½ ft. wide, 6 in. thick on top, and 5 in. at the bottom amidships, tapering to 3½ in. at the ends. Above the belt, for the length of the battery, they have 5 in. armour. Their guns are worked in pairs in turrets, having 6½ in. armour with 6 in. barbette. The *California*, *Nebraska*, and *West Virginia*, of this class, are sheathed, the *Maryland*, *Colorado*, *South Dakota* are unsheathed. The unsheathed vessels are about 700 tons less displacement than the *Drake* class (which are also unsheathed), and they have a knot less speed, and 300 tons less coal, but 1 ft. 6 in. less draught. The *Drake* class has two 9·2 in. guns, against four 8 in. guns, and a casemated battery against a box battery. It is very difficult to compare ships of different classes with each other, and even ships of the same class in different navies. For instance, *Maryland* is an unsheathed cruiser, but closely approximates to the new U.S. battleship in the character of the arrangement of armour and armament, but in the power of these qualities she is necessarily inferior. Their relative speeds on trial are 22 and 19 knots, and, probably, in ordinary working condition for long periods would be 19½ and 17½ knots. The gun power of the battleship is overwhelmingly greater than the cruiser, and she has only her 2 knots sea speed, or 3 knots trial speed, to compensate. The cost of the cruiser is 10 per cent. more than the battleship. She has a crew 18 per cent. larger, probably in consequence of her higher speed. It is difficult to estimate the value of two to three extra knots speed, but it may be seen from this direct comparison what it entails in the form of armour, armament, first cost, and upkeep expenses.

If we attempt to compare *Drake* and *Maryland*, we have again to value a difference in speed of 1 knot and 25 per cent. more coal against the difference between two 9·2 guns and four 8 in. guns. Here, again, the Americans seem to prefer a dominating armament to other qualities. It is, at any rate, a quality which is more

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likely to have a real existence in a ship in time of battle, as speed is dependent upon the state of the bottom and the efficiency of the machinery, and is very apt to be less than it is expected to be.

The *St. Louis* class is of 9,700 tons displacement and 22 knots speed, with a normal supply of 650 tons coal. Each ship has a partial belt amidships, 200 ft. long, 7½ ft. wide, and 4 in. thick, surmounted by a partial belt, 133 ft. long and 4 in. thick, extending to the upper deck. The protective deck behind the armour is 3 in. on the slope and 2 in. on the flat. The principal armament consists of fourteen 6 in. guns, eight in a box battery, four mounted on the upper deck, immediately over the corner guns in the battery below, and protected in front by 4 in. armour, and two mounted in the open at the middle line, fore and aft on the upper deck. The following Table gives the thicknesses and extent of armour in the principal ships.

TABLE A.

| ARMOUR. | *Arkansas.* | *Iowa.* | | *Kentucky.* | | *Alabama.* | *Maine.* | *Virginia and Georgia.* | | *California.* | *St. Louis.* |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Length of W.L. belt ... | Stem to stern | 180′ 0″ | | 260′ 0″ | | 220′ 0″ | 260′ 0″ | Stem to stern | | Stem to stern | 200′ amidship |
| Total depth of W.L belt ... | 5′ 0″ | 7′ 6″ | | 7′ 6″ | | 7′ 6″ | 7′ 6″ | 8′ 0″ | | 7′ 6″ | 7′ 6″ |
| Thickness midships top .. | 11″ | 14″ | | 16½″ | | 16½″ | 12″ | 11″ | | 6″ | 4″ |
| Thickness midships bottom | 5″ | 7″ | | 9½″ | | 9½″ | 8½″ | 8″ | | 5″ | 4″ |
| Length midship thickness... | 80′ | 180′ | | 140′ | | 140′ | 140′ | About 180′ | | 244′ | 200′ |
| Thickness at ends ... ... | 5″ top, 3″ bot. | — | | 4″ at stem | | 4″ at stem | 4″ at stem | 4″ | | 3½″ | — |
| Length lower to maindk. belt | — | 120′ | | 200′ | | 140′ | 160′ | 245′ | | 232′ | 133′ |
| Thickness " " | — | 5″ | | 5″ | | 5½″ | 5½″ | 6″ | | 5″ | 4″ |
| Length main to upper dk. belt | — | — | | 120′ | | 140′ | 140′ | 245′ | | 232′ | 133′ |
| Thickness " " | — | 2″ | | 6″ | | 5½″ | 6″ | 6″ | | 5″ | 4″ |
| Thickness transverse armour bulkhead ... .. ... | — | 12″ | | 12″ A. & 10″ F. | | 12″ | 12″ | 6″ inclined | | 4″ | 3″ |
| | | 13″ guns | 8″ guns | 13″ guns | 8″ guns | | | 12″ guns | 8″ guns | | |
| Turrets max. thickness ... | 10″ | 15″ | 6″ | 15″ | 11″ | 17″ | 17″ | 11″ | 8″ | 6½″ | — |
| Turrets min. thickness ... | 10″ | 15″ | 6″ | 12″ | 9″ | 15″ | 15″ | 9″ | 6½″ | 6″ | — |
| Barbettes max. thickness... | 11″ | 15″ | 8″ | 15″ | | 15″ | 15″ | 10″ | 6″ | 6″ | — |
| Barbettes min. thickness... | 11″ | 15″ | 6″ | 15″ | | 10″ | 10″ | 6″ | 4″ | 6″ | — |
| Protective decks flats ... | 1½″ | 2¾″ | | 2¾″ | | 2¾″ | 2¾″ | 1½″ | | 1½″ | 2″ |
| Protective decks slopes ... | — | — | | 5″ | | 4″ | 4″ | 3″ | | 4″ | 3″ |



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The vessels of the *St. Louis* class are very similar to our County class, but they have a knot less speed. Our guns are the same in number and calibre, ten of them being in casemates, six on the main, and four on the upper deck, where the remainder are mounted in two pairs in turrets 5 in. thick at the middle line, forward and aft. The normal coal supply of our ships is 150 tons more than the United States ships. Our ships seem to be at least equal to the United States ships of this class; but it should be noted that the superiority in speed of one knot is expected to be obtained with practically the same horse-power. The relative value in speed cannot, therefore, be properly judged until the vessels are tried, as the United States builders will probably provide for a greater margin of speed that we may expect in our ships, as they guarantee the speed, and not the power, as our builders do.

The ships of the *Denver* class are 3,200 tons displacement, steam 16½ knots, and carry 470 tons coal. They have no side armour, but have each a protective deck, 2 in. on the slope, ½ in. on the flat. Their principal armament consists of ten 5 in. guns on the upper deck. They are sheathed. These vessels are like our small second-class cruisers; but they are nominally 3 knots slower on trial, and do not seem to be any improvement upon our vessels. There are no vessels in the United States Navy of a class similar to our third-class cruisers; but there have been built, since 1891, nine gunboats of 1,000 to 1,300 tons, and of 13 to 15 knots speed. They have no protection, their principal armament being six 4 in. guns. They have a very limited amount of subdivision, and can hardly be looked upon as war vessels of much power, but act as good training ships.

In the torpedo boat and destroyer classes, the types are varied. The United States Naval Constructors began by making vessels similar to those of the European types, but they have departed from these, and developed vessels of a slower but more powerful description. Up to 1894, three small torpedo boats were all that the United States Navy possessed, but in that year three torpedo boats of 144 tons and 24½ knots speed were ordered. In 1895, three more of 165 tons, and one of 180 tons were ordered. The former obtained 28½ knots, the latter 27 knots speed. In 1896, ten boats were ordered. They were: one a destroyer, the *Farragut*, of 240 tons and 30 knots speed; two, the *Dahlgren* and the *Craven*, of 144 tons and 30 knots; two 110 ton boats of 22½ knots; one of 98 tons and of 22½ knots; two of 65 tons and 20 knots, and two of 45 tons and 20 knots. The *Farragut* was of the *Desperate* type, and the *Dahlgren* and the *Craven* were similar to the *Cyclone*, built by Normand for the French Government. In 1897 three destroyers were ordered. They were all of 30 knots speed, one of 340 tons, one of 248 tons, and one of 235 tons.

In 1898, twelve torpedo boats of about 165 tons and 26 knots were ordered, but in three of them the builders guaranteed 28 knots. There were also sixteen destroyers of 400 tons to 420 tons ordered. There were two of 28 knots, nine of 29 knots, and

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five of 30 knots. Since 1898 more have been ordered. The plans of the *Rowan*, of 184 tons, and *Farragut*, of 240 tons, are given, and also the department plans for the 400 tons destroyers are shown. These last vessels were 245 ft. by 23½ ft. by 6½ ft., with an armament of two 12-pounders, five 6-pounders, and two 18 in. torpedo tubes in the broadside. Their load on trial is 46 tons.

TABLE G.

TABLE OF COMPARISON. SHIPS OF 1891. PROPOSED AND ACTUAL DATA.

| — | | Displacement. | Draught. | Maximum I.H.P. | Speed. | Revolutions. | Normal Coal Capacity | Total Bunker Capacity. | Steaming Distance at 10 Knots. |
|---|---|---|---|---|---|---|---|---|---|
| | | Tons. | Ft. in. | | Knots. | | Tons. | Tons. | |
| *New York* .. | Proposed | 8,150 | 23 3½ | 16,500 | 20 | 129 | 750 | 1,500 | 13,500 |
| | Actual... | 8,200 | 23 3½ | 17,401 | 21 | 135 | 750 | 1,290 | 5,000 |
| *Olympia* ... ... | Proposed | 5,500 | 21 6 | 13,500 | 20 | 129 | 400 | 1,300 | 13,000 |
| | Actual... | 5,870 | 21 6 | 17,313 | 21·68 | 140 | 400 | 1,169·6 | 6,105 |
| *Raleigh* ... ... | Proposed | 3,983 | 18 0 | 10,000 | 19 | — | — | — | — |
| | Actual... | 3,213 | 18 0 | — | — | — | 350 | 460·0 | 2,940 |
| *Cincinnati* ... | Actual... | 3,213 | 18 0 | — | — | — | 384 | 577 | 3,000 |
| *Detroit* ... ... | Proposed | 2,000 | 14 6 | 5,400 | 17 | — | 200 | — | — |
| | Actual... | 2,089 | 14 7 | 5,227 | 18·71 | 13 | 200 | 340 | 3,280 |
| *Montgomery* ... | Actual... | 2,089 | 14 7 | 5,580 | 19·05 | — | 200 | 340 | 3,200 |
| *Marblehead* ... | Actual... | 2,089 | 14 7 | 5,451 | 18·44 | 176 | 200 | 340 | 2,900 |
| *Columbia*... ... | Proposed | 7,475 | 24 0 | 21,000 | 21 | 129 | 750 | 2,000 | 26,240 |
| | Actual... | 7,375 | 22 6 | 18,509 | 22·8 | 132 | 750 | 1,670 | 7,200 |
| *Minneapolis* ... | Actual... | 7,375 | 22 6 | 20,862 | 23·07 | 132 | 750 | 1,891 | 6,300 |
| *Indiana* ... ... | Proposed | 10,200 | 24 0 | 9,000 | 15 | 128 | 400 | 1,800 | 16,000 |
| | Actual... | 10,288 | 24 0 | 9,738 | 15·54 | 131 | 400 | 1,597 | 3,720 |
| *Massachusetts* ... | Actual... | 10,288 | 24 0 | 10,415 | 16·21 | 133 | 400 | 1,597 | 4,500 |
| *Oregon* ... ... | Actual... | 10,288 | 24 0 | 11,111 | 16·79 | 128 | 400 | 1,594 | 5,500 |
| *Katahdin* .. ... | Proposed | 6,750 | 8 4 | 6,000 | 23 | 333 | — | — | — |
| | Actual... | 2,155 | 15 0 | 5,068 | 16·11 | 145 | 175 | 192·7 | 1,000 |

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Having described briefly the ships proposed since 1891, it may be desirable to notice the results obtained in the ships referred to in my paper of 1891. Table B includes the results of these ships as finished. The items, which can only be determined when the ship is completed, are given in Table G, alongside the corresponding figures for the ship as proposed.

Practically they all carried their weights on the designed draught, the *New York* being the only notable exception. The speed was generally exceeded by 1 to 2 knots, the *Katahdin* being the only notable exception; but she was of a very unusual form, proposed by Admiral Ammon. The amount of bunker space was generally less than designed, but this is not an uncommon experience, as, where bunker space is large, it is freely encroached upon for many purposes during the construction of a ship. The most serious shortcoming is in the steaming distances, but these were promised in the days when we were all much too sanguine in this respect, and before many vessels had been submitted to practical trials of the distance they could steam.

It may be interesting to compare the proposed and actual figures of later ships which have been completed.

TABLE H.

TABLE OF COMPARISON. SHIPS AUTHORISED AND COMPLETED AFTER 1891.

| — | | Displacement. | Draught. | Maximum I.H.P. | Speed. | Revolutions. | Normal Coal Capacity | Total Bunker Capacity. | Steaming Distance at 10 Knots. |
|---|---|---|---|---|---|---|---|---|---|
| | | Tons. | Ft. in. | | Knots. | | Tons. | Tons. | |
| *Iowa* ... ... | Proposed | — | 24 0 | 10,000 | 16 | 100 | — | — | — |
| | Actual... | 11,340 | 24 0 | 12,105 | 17·103 | 110 | 625 | 1,795 | 4,000 |
| *Kentucky* ... ... | Proposed | 11,500 | 23 6 | 10,000 | 16 | 110 | — | — | — |
| | Actual... | 11,540 | 23 6 | 12,318 | 16·897 | 114 | 410 | 1,591 | — |
| *Kearsage* ... ... | Actual... | 11,540 | 23 6 | 11,954 | 16·816 | 114 | 410 | 1,591 | — |
| *Alabama* ... ... | Proposed | 11,520 | 23 6 | 10,000 | 16 | 110 | 800 | 144 | 1,355 |
| | Actual... | 11,570 | — | 11,366 | 17·013 | 115 | — | — | — |
| *Wisconsin* ... | Actual... | — | — | — | 17·174 | 119 | — | — | — |
| *Brooklyn* ... ... | Proposed | 8,150 | 23 3½ | 16,500 | 20 | 129 | 750 | 1,500 | 13,500 |
| | Actual... | 9,215 | 24 0 | 18,709 | 21·91 | 136 | 900 | 1,461 | 5,110 |

From the foregoing particulars it may be seen that the U.S. Naval Constructors have produced ships which should give them satisfaction in their results. Their

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cruisers generally call for little comment. Ours are not very dissimilar, and will, it is hoped, when finished, give satisfaction to their designer. The most striking difference is in their battleships. The differences are of two kinds. *First*, the protection of the 6 in. battery, which is armoured all round. *Second*, the addition of eight 8 in. protected guns to the principal armament. The former has been already discussed, but the latter deserves a few more remarks. The system of carrying four of these guns in turrets superposed upon the turrets of the 12 in. guns has been advocated by the Ordnance Bureau, and opposed by the Construction Bureau. The arguments in favour may best be stated in the words of Admiral O'Neil, Chief of the Ordnance Bureau. He says:—

"During the past year the battleships *Kearsage* and *Kentucky* have been completed and put in commission. Much attention has been attracted to these vessels, both at home and abroad, on account of the novel arrangement of their heavy guns, four 8 in. guns on each vessel being mounted in two turrets placed on top of, and rigidly attached to, the turrets containing the 13 in. guns.

"These vessels successfully stood their gun trials, and the double turret structures, which are trained by electric power, proved to be under perfect control, and showed no signs of weakness, nor was any inconvenience experienced in the upper or lower turret due to the firing of the guns above or below. In fact, no unfavourable reports concerning the arrangement of the battery have been received.

"The vessels are, without doubt, an unqualified success, and, while there is and probably always will be a decided difference of opinion among naval officers as to the merits, or, perhaps, more properly speaking, as to the advisability of the system, it must be admitted that it possesses certain very attractive features, among which are an absolute non-interference of guns, a heavy and unobstructed bow and stern fire, and the fact that the 8 in. guns—being mounted on the line of the keel—are available for service on either side, resulting in a very considerable saving of weight for equal efficacy.

"The arguments for and against superimposed turrets are well understood, and need not be repeated, but it is a fact worthy of note that a considerable number of naval men, who were opposed to the superimposed turrets for various reasons, have changed their views since they have seen them completed and in service."

The arguments against the system, which appear to be stated as a reply to the Ordnance Bureau, are given by Admiral Hichborne, the chief of the Bureau of Construction, who says:—"The completion of the first battleships to which the system has been applied, and the trials to which they have been subjected have demonstrated only the mechanical practicability of the system of mounting. It has been recognised from the commencement of the discussion that there are such

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grave disadvantages in this system of mounting as: impossibility of independent action of 8 in. guns in superposed turrets; great concentration of weight upon roller path and supports of such turrets; dependence of four important guns upon one controlling apparatus; disarrangement of training of three guns by firing of the fourth, and possibility of complete disablement of four important guns, or, say, 35 per cent. of the heavy battery, by mishap affecting the turret. There is nothing in the experience with the two vessels recently completed tending either to show how they may be removed, or to determine the extent of their effect upon the vessels in which this system is employed. The decision in the case of the recent vessels has been reached in practically the same way as the decision relative to the first vessels to which the system was applied, and is based upon the same representation of arguments *pro* and *con.*, although the Navy now possesses the means of determining, practically, at any rate a portion of the points at issue. Other naval Powers are pursuing a policy of isolation of the larger pieces in single-gun, or, at most, two-gun positions (turrets, enclosed casemates, &c.), and making them independent of and without communication with one another, with the avowed purpose of limiting both the material damage to the individual pieces, and the moral effect to the single crew.

"The Bureau believes the arrangement of battery of vessel of war to be a matter of the very greatest importance. While seriously opposed to the system of superposed turrets, on grounds which it believes to be vital, it holds itself free from any prejudice which could not be removed by practical tests which the means now at hand make possible, and which the importance of the matter makes imperative, in order to limit a policy of compromise of opinions based upon theoretical rather than upon practical considerations."

Comparing the arrangement of the *Virginia* with the *Georgia*, as first proposed, it may be remarked that the arrangement in the *Indiana*, which is very similar to the *Georgia*, has not given complete satisfaction, on account of the interference of the 8 in. and the 12 in. guns. It is almost impossible, when firing ahead with the forward 8 in. guns, to work in the forward 12 in. turrets. This is avoided by the superposed turret, which for two guns has as much power of broadside fire in one broadside as the four 8 in. It would seem from this that the fitting of the two independent 8 in. turrets in the *Virginia* in addition to the superposed 8 in. turrets, is repeating the difficulty (at any rate for one end of the ship) which has been experienced in the *Indiana* class, and, in consequence, a logical conclusion would be to do away with these independent turrets, and use the weight in another way.

Whichever arrangement is adopted, the fact remains that these ships will have eight 8 in. guns, or its equivalent weight of armament of 400 tons more than our battleships. Their armour protection is thicker and more extended, as they have a

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complete belt, against a belt about five-sixths the length of our battleships. They are to have one knot more speed.

Some rather important details, which are common to the larger classes of American ships, are worth noticing. Cofferdams filled with obturating material, which is expected to expand when in contact with water, are fitted very generally at the sides of the ship. This material is the pith of the corn stock, and has been experimented upon very fully by the Navy Department, with the result that they prepare their designs with the intention of adopting it generally. It is evident that if the corn stock material swells when in contact with water sufficiently to fill up holes made by shot, it will have an important effect upon the margin of stability and probably of buoyancy of a ship in action.

The watertight doors throughout the ship can be closed from a central station under the control of one officer. This has often been advocated, but has not been adopted as a general system in any navy until quite recently.

It is the general practice in the United States battleships to adopt what they call docking bilge keels. These side keels act as bilge keels, and also act as side keels for docking, distributing the pressure over a larger area of the bottom of the dock and reducing the stress upon the structure of the ship.

It may be noticed that the complements of the United States Navy ships are generally less than those of our ships.

| | *Virginia.* | *Maryland.* | *St. Louis.* |
|---|---|---|---|
| | Tons. | Tons. | Tons. |
| Weight of hull ... ... ... ... ... | 6,652 } 10,342 | 6,700 } 8,919 | 4,912 } 5,766 |
| Weight of protection ... ... ... | 3,690 | 2,219 | 854 |
| Weight of equipment ... ... ... | 663 | 675 | 592 |
| Weight of guns and ammunition ... (Two-thirds full supply.) | 1,213 | 696 | 642 |
| Weight of machinery ... ... ... | 1,730 | 2,060 | 1,900 |
| Weight of F. W. for machinery ... | 100 | 150 | 150 |
| Weight of coal (normal) ... ... ... | 900 | 900 | 650 |
| | 14,948 | 13,400 | 9,700 |
| Block co-efficients ... ... ... ... | ·661 | ·561 | ·5 |

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It is undoubted that in the battleships the 8 in. guns would be able to penetrate the 6 in. protection of our casemates, and as with the *Virginia* arrangement they can fire six on each broadside, there can be little doubt of the great value of these additional guns. If it is decided to retain the 8 in. guns in the independent positions, it would make, in some respects, a better arrangement if they were in echelon, as then they would all fire on each broadside, and could do as well as at present in right-ahead or right-astern positions.

It may be interesting to put in tabular form some of the principal weights of the three new classes, the battleship *Virginia*, the armoured cruiser *Maryland*, and the protected cruiser *St. Louis*. This is done on the previous page.

These block co-efficients appear to be higher than those adopted in European battleships, and to approach merchant ship practice. The *Formidable*, *Drake*, and *Essex*, being respectively ·654, ·534, ·482; but in the later type of British battleship, the *Duncan*, a block co-efficient as low as ·604 has been adopted. This low co-efficient means a sacrifice of 1,400 tons (10 per cent. of displacement), compared with the *Virginia* class. This must have been due to a very strong reason, and be associated with a great countervailing advantage which is not very apparent, as the speeds and horse-powers are nominally the same.

NOTE.—With reference to the steaming distances of which these ships are capable, a very interesting tabular collection of data on this subject has recently been published by Lieutenant E. C. Bryan, U.S.N., in the Journal of the American Society of Naval Engineers. From these tables I have extracted, for some of the best ships, the distances which they are capable of steaming at different speeds, and the conditions in which they were tried. These results are shown in diagram form (Plate XXXIII.), and, though they are not very consistent, on account of the varying conditions under which the trials were made, they show the results a little more clearly in this form than in the tables. *Columbia*, for instance, gives comparatively a fair curve through all the spots, which shows that at a most economical speed, which seems to be something between 12 to 15 knots, she cannot steam more than 5,000 knots.

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| | | *Texas.* | *Maine.* | *Monterey.* | *Cushing.* |
|---|---|---|---|---|---|
| Length on | n. | 301 ft. 0 in. | 318 ft. 0 in. | 256 ft. 0 in. | 137 ft. 6 in. |
| Breadth | n. | 64 ft. 1 in. | 57 ft. 3 in. | 59 ft. 0 in. | 14 ft. 3 in. |
| Mean drau | n. | 22 ft. 6 in. | 21 ft. 6 in. | 14 ft. 10 in. | 4 ft. 10½ in. |
| Displacem | | 6,315 | 6,682 | 4,084 | 1,053 |
| Maximum | | 8,610 | 9,293 | 5,244 | 1,750 |
| Speed | | 17·8 | 17·45 | 13·6 | 22·53 |
| Armament | F. | 2 12 in. B.L.R.<br>6 6 in. B.L.R. | 4 10 in. B.L.R.<br>6 6 in. B.L.R. | 2 12 in. B.L.R.<br>2 10 in. B.L.R. | 3 1-pdrs. |
| | | 25 small guns | 19 small guns | 7 small guns | |
| Torpedo tu | | 2 | 4 | 0 | 2 |
| Engines | .<br>56 | Two sets.<br>36 × 51 × 78<br>39 in. | Two sets.<br>35½ × 57 × 88<br>36 in. | Two sets.<br>27 × 41 × 64<br>30 in. | Two sets.<br>11½ × 16 × 22½ × 22½<br>15 in. |
| Cost in dol | | 2,500,000 | 2,500,000 | 1,628,950 | 82,750 |
| Revolution | | 125 | 124 | 161 | 385 |
| Piston spee | | 812 | 744 | 805 | 935 |
| Heating su | | 16,912 | 19,015·3 | 14,785 | 2,252 |
| Grate surfa | | 531·6 | 573·8 | 383 | 38·6 |
| Working pl | | 150 | 135 | 160 | 235 |
| Normal co | | 500 | 400 | 200 | 10 |
| Total bunk | | 850 | 896 | 233 | 36 |
| Endurance | | 2,900 | | 1,430 | 1,092 (at 14 knots) |

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TABLE C.—BATTLESHIPS.

| — | *Iowa.* | *Kentucky. Kearsage.* | *Alabama. Illinois. Wisconsin.* | *Maine. Missouri. Ohio.* | *New Jersey. Georgia. Pennyslvania.* | *Virginia. Rhode Island.* | *Arkansas. Nevada. Florida. Wyoming.* |
|---|---|---|---|---|---|---|---|
| Average contract cost (exclusive of armour) | $3,010,000 | $2,250,000 | $2,639,300 | $2,889,600 | $3,640,000 | $3,405,000 | $860,000 |
| Date of authorisation | 1892 | 1895 | 1896 | 1898 | 1899 | 1900 | 1898 |
| Class ... ... ... | Sea-going B. | Sea-going B. | Sea-going B. | Sea-going B. | Sea-going B. | Sea-going B. | Coast Defence Monitors |
| Length L.W.L. ... | 360′ 0″ | 368′ 0″ | 368′ 0″ | 388′ 0″ | 435′ 0″ | 435′ 0″ | 252′ 0″ |
| Length overall ... | 362′ 6″ | 376′ 0″ | 373′ 9″ | 393′ 9″ | 441′ 4″ | 441′ 4″ | 258′ 0″ |
| Breadth (moulded)... | 72′ 0″ | 72′ 0″ | 72′ 0″ | 72′ 0″ | 76′ 0″ | 76′ 0″ | 49′ 10″ |
| Breadth (extreme)... | 72′ 2½″ | 72′ 2½″ | 72′ 2½″ | 72′ 2½″ | 76′ 10″ | 76′ 2½″ | 50′ 0″ |
| Freeboard amidships | 15′ 0″ | 11′ 0″ | 19′ 10″ | 20′ 3½″ | 18′ 3″ | 18′ 3″ | 2′ 6¾″ |
| Mean draught ... | 24′ 0″ | 23′ 6″ | 23′ 6″ | 23′ 6″ | 24′ 0″ | 24′ 0″ | 12′ 6″ |
| Displacement ... | 11,340 | 11,540 | *a*11,565 | *a*12,400 | *a*15,320 | *a*14,948 | *a*3,235 |
| Maximum I.H.P. ... | 12,105 | 12,180 | *a*10,000 | *a*16,000 | *a*19,000 | *a*19,000 | *a*2,400 |
| Speed ... ... ... | 17·103 | 16·85 | *a*16·00 | *a*18 | *a*19 | *a*19 | *a*11·5 |
| Area mids. sect. (sq.ft.) | 1,637 | 1,623 | 1,613 | 1,637 | 1,745 | 1,730 | 594 |
| Block co-efficient ... | ·638 | ·646 | ·637 | ·655 | ·661 | ·661 | ·74 |
| Armour — Midship thickness | W.L. Belt 14″ | 16½″ top, 9½″ bot. | 16½″ top, 9½″ bot. | 11″ top, 7½″ bot. | 11″ top, 8″ bot. | 11″ top, 8″ bot. | 11″ |
| Armour — 13″ or 12″ guns | Turrets 15″ | 17″ | 14″ | 12″ | 11″ | 11″ | 10″ |
| Armour — 8″ guns | Turrets 8″ | — | — | — | 6½″ | 6½″ | — |
| Armour — 13″ or 12″ guns | Barbettes 15″ | 15″ | 15″ | 12″ | 10″ | 10″ | 11″ |
| Armament ... ... | 4 12″ B.L.R.<br>8 8″ B.L.R.<br>6 4″ R.F.<br>30 Small | 4 13″ B.L.R.<br>4 8″ B.L.R<br>14 5″ R.F.<br>34 Small | 4 13″ B.L.R.<br>14 6″ B.L R.<br>—<br>28 Small | 4 12″ B.L.R.<br>16 6″ R F.<br>—<br>24 Small | 4 12″ B.L.R.<br>8 8″ B.L.R.<br>12 6″ R.F.<br>42 Small | 4 12″ B.L.R.<br>8 8″ B.L.R.<br>12 6″ R.F.<br>42 Small | 2 12″ B.L.R.<br>4 4″ R.F.<br>—<br>10 Small |
| Torpedo tubes ... | Four | Four | Four | Two | Two | Two | — |
| Engines ... ... | Two sets<br>39 × 55 × 85 / 48″ | Two sets<br>33½ × 51 × 78 / 48″ | Two sets<br>33½ × 51 × 78 / 48″ | Two sets<br>38½ × 59 × 92 / 42″ | Two sets<br>35 × 57 × 66 / 48″ | Two sets<br>35 × 57 × 66 / 48″ | Two sets<br>17 × 26¼ × 40 / 24″ |
| Revolutions ... ... | 110 | 114 | 120 | 126 | 120 | 120 | 200 |
| Boilers ... ... | Cylindrical | Cylindrical | Cylindrical | Water-tube | B. & W. | B. & W. | Water-tube |
| Heating surface (sq. ft.) | 24,082 | 22,104 | 21,205 | 58,104 | 55,000 | 55,000 | 8,800 |
| Grate surface (sq. ft.) | 756 | 685 | 640 | 1,353 | 1,280 | 1,280 | 200 |
| Working pressure ... | 160 | 180 | 180 | 250 | 250 | 250 | 250 |
| Normal coal cap. (tons) | 625 | 410 | 800 | 1,000 | 900 | 900 | 200 |
| Total bunker cap. (tons) | 1,795 | 1,591 | 355 | 2,000 | 1,900 | 1,900 | 400 |
| Endurance at 10 knots | 4,000 | — | — | — | — | — | — |

(*a*) Estimated.

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TABLE D.—ARMOURED CRUISERS.

| — | *California. Nebraska. West Virginia* (sheathed). | *Maryland. Colorado. South Dakota.* | *St. Louis. Milwaukee. Charleston.* | *Denver. Des Moines. Tacoma. Chattanooga. Galveston. Cleveland* (sheathed). |
|---|---|---|---|---|
| Class ... ... ... ... ... | 1st class armoured cruiser. | 1st class armoured cruiser. | 1st class protected cruiser. | Protected cruiser. |
| Date of authorisation ... ... | 1899 | 1900 | 1900 | 1899 |
| Average contract cost (exclusive of armour.) ... ... | $3,858,300 | $3,768,000 | $2,743,700 | $1,049,250 |
| Length L.W.L. ... ... ... | 502 ft. 0 in. | 502 ft. 0 in. | 424 ft. 0 in. | 292 ft. 0 in. |
| Length overall ... ... ... | 504 ft. 9 in. | 504 ft. 9 in. | 426 ft. 6 in. | 308 ft. 9 in. |
| Breadth (moulded) ... ... | 69 ft. 4 in. | 69 ft. 4 in. | 65 ft. 2 in. | 43 ft. 3¼ in. |
| Breadth (extreme) ... ... | 70 ft. 0 in. | 69 ft. 6 in. | 66 ft. 0 in. | 44 ft. 0 in. |
| Freeboard amidships ... ... | 18 ft. 0 in. | 18 ft. 0 in. | 17 ft. 8 in. | 15 ft. 9 in. |
| Mean draught ... ... ... | 24 ft. 6 in. | 24 ft. 6 in. | 23 ft. 6 in. | 15 ft. 9 in. |
| Displacement ... ... (tons) | 13,800 | 13,400 | 9,700 | 3,200 |
| Maximum I.H.P. ... ... ... | 23,000 | 23,000 | 21,000 | 4,500 |
| Speed ... ... (knots) | 22 | 22 | 22 | 16·5 |
| Area midship section ... ... | 1,607·2 | 1,575·0 | 1,445·6 | 615·2 |
| Block co-efficient ... ... ... | ·561 | ·561 | ·5 | — |
| Armour belt ... ... ... | Complete belt.<br>6 in. top.<br>5 in. bottom.<br>3½ in. ends. | Complete belt.<br>6 in.<br>5 in.<br>3½ in. | Part. belt (200 ft. long)<br>4 in.<br>4 in.<br>— | —<br>None.<br>None.<br>None. |
| Battery protection ... ... | 5 in. | 5 in. | 4 in. | — |
| Armament ... ... ... ... | 4 8 in. B.L.R.<br>14 6 in. B.L.R.<br>40 small guns. | 4 8 in. B.L.R.<br>14 6 in. B.L.R.<br>40 small guns. | 14 6 in.<br>—<br>40 small guns. | 10 5 in.<br>—<br>14 small guns. |
| Torpedo tubes ... ... ... | 2 | 2 | 0 | 0 |
| Engines ... ... ... | Two sets.<br>38½ × 63½ × 74, 74<br>48 in. | Two sets.<br>38½ × 63½ × 74, 74<br>48 in. | —<br>—<br>48 in. | Two sets.<br>18 × 29 × 35½, 35½<br>54 in. |
| Revolutions ... ... ... | 120 | 120 | 125 | — |
| Boilers ... ... ... ... | Babcock & Wilcox. | Babcock & Wilcox. | Babcock & Wilcox. | 6 water-tube. |
| Heating surface ... (sq. ft.) | 68,000 | 68,000 | 58,800 | 12,600 |
| Grate surface ... (sq. ft.) | 1,590 | 1,590 | 1,440 | 314 |
| Working pressure... (lbs. sq. in.) | 250 | 250 | 250 | 250 |
| Normal coal ... ... (tons) | 900 | 900 | 650 | 467 |
| Total bunker capacity ... (tons) | 2,000 | 2,000 | 1,500 | 700 |
| Steaming distance (at 16 knots) ... | — | — | — | 7,000 knots (estimated.) |

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| Tingey. | Wilkes. | Bainbridge. Barry. Chauncey. | Dale. Decatur. | Hopkins. Hull. | Lawrence. Macdonough. | Paul Jones. Perry. Preble. | Stewart. | Truxtun. Whipple. Worden. |
|---|---|---|---|---|---|---|---|---|
| 175′ 0″ | 175′ 0″ | 245′ 0″ | 245′ 0″ | 244′ 0″ | 240′ 10″ | 245′ 0″ | 245′ 0″ | 248′ 0″ |
| 17′ 6″ | 17′ 7⅝″ | 23′ 7¼″ | 23′ 7¼″ | 24′ 6″ | 22′ 3″ | 23′ 7¼″ | 23′ 7¼″ | 23′ 3″ |
| 4′ 8″ | 4′ 8″ | 6′ 6″ | 6′ 6″ | 6′ 0″ | 6′ 1½″ | 6′ 6″ | 6′ 6″ | 6′ 0″ |
| 165 | 165 | 420 | 420 | 408 | 400 | 420 | 420 | 433 |
| — | — | 535 | 535 | 537 | 489 | — | 535 | 610 |
| (e) 7′ 6″ | (e) 7′ 6″ | (e) 7′ 6¾″ | (e) 7′ 6¾″ | -- | — | (e) 7′ 6¾″ | — | — |
| 3 3-pdr. | 3 3-pdr. | 2 3″ R.F. 5 6-pdr. | 2 3″ R.F. 5 6-pdr. | 2 3″ R.F. 5 6-pdr. | 2 3″ R.F. 5 6-pdr. | 2 3″ R.F. 5 6 pdr. | 2 3″ R.F. 5 6-pdr. | 2 3″ R.F. 5 6-pdr. |
| 3 | 3 | 2 | 2 | 2 | 2 | 2 | 2 | 2 |
| (e) 26 | (e) 26½ | (e) 29 | (e) 28 | (e) 29 | (e) 30 | (e) 29 | (e) 29 | (e) 30 |
| (e) 3,000 | (e) 3,000 | (e) 8,000 | (e) 8,000 | (e) 7,200 | (e) 8,400 | (e) 7,000 | (e) 8,000 | (e) 8,300 |
| — | — | 174 | 174 | — | — | 174 | 174 | — |
| (e) 80 | (e) 80 | (e) 189 | (e) 189 | (e) 189 | (e) 189 | (e) 189 | (e) 189 | (e) 189 |
| — | — | 46 | 46 | 46 | — | 46 | 46 | 46 |
| 10 | 10 | 25 | 25 | 25 | 34 | 25 | 25 | 25 |
| (e) 70 | (e) 70 | 139 | 139 | 150 | 122·6 | — | 139 | 202 |
| 4×22×25¼, 25¼ / 18″ | 14×22×25½, 25½ / 18″ | 20½×32×33, 38 / 22″ | 20½×32×38, 38 / 22″ | 22×32½×34, 34 / 22″ | 22×31×34, 34 / 20″ | 20½×32×38, 38 / 22″ | 20½×32×38, 38 / 22″ | 23×34×37, 37 / 20″ |
| — | — | — | — | — | — | — | — | — |
| 3 Thorny. | 3 Seabury | 4 Thorny. | 4 Thorny. | 4 Thorny. | 4 W.T. | 4 Thorny. | 4 Thorny. | 4 Normand |
| 137 | 137·4 | 315 | 315 | 294 | 304 | 315 | 315 | 300 |
| 7,548 | 7,800 | 17,768 | 17,768 | 17,612 | 18,117 | 17,768 | 17,768 | 19,748 |
| — | — | — | — | — | — | — | — | — |

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TABLE E.—GUNBOATS.

| GUNBOATS. | Length. | Breadth. | Draught. | Displacement. | Type of Engine. | Speed. | Maximum I.H.P. | Tons per Inch. | Normal Coal Supply. | Bunker Capacity. | Batteries. | Complement. Officers. | Complement. Men. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ft. in. | Ft. in. | Ft. in. | Tons. | | | | | Tons. | Tons. | | | |
| *Annapolis* ... | 168 0 | 36 0 | 12 0 | 1,000 | S.S. Vert. Trip. | 13·17 | 1,227 | 10·75 | 100 | 225 | 6 4 in. R.F.<br>4 6-pdr. R.F.<br>2 1-pdr. R.F.<br>1 Colt<br>2 Field gun. | 11 | 124 |
| *Marietta* ... | 174 0 | 34 0 | 12 0 | 1,000 | T.S. Vert. ,, | 13·03 | 1,054 | 10 17 | 120 | 120 | do. | 11 | 129 |
| *Newport* ... | 168 0 | 36 0 | 12 0 | 1,000 | S.S. Vert. ,, | 12·29 | 1,068 | 10 75 | 100 | 100 | do. | 11 | 124 |
| *Princetown* .. | 168 0 | 36 0 | 12 0 | 1,000 | S.S. Vert. ,, | 12·0 | 800 | 10 75 | 100 | 100 | do. | 11 | 124 |
| *Vicksburg* ... | 168 0 | 36 0 | 12 0 | 1,000 | S.S. Vert. ,, | 12·71 | 1,118 | 10·75 | 100 | 100 | do. | 11 | 124 |
| *Wheeling* ... | 174 0 | 34 0 | 12 0 | 1,000 | T.S. Vert. ,, | 12·88 | 1,081 | 10 17 | 120 | 120 | do. | 11 | 129 |
| *Helena* ... | 250 9 | 40 0 11/16 | 9 0 | 1,392 | T.S. Vert. ,, | 15·5 | 1,988 | 17·08 | 100 | 300 | 8 4 in. R.F.<br>4 6-pdr.<br>4 1-pdr.<br>2 Colts (1 Field gun). | 10 | 165 |
| *Nashville* ... | 220 0 | 38 1½ | 11 0 | 1,371 | T.S. Quad. | 16·3 | 2,536 | 13·16 | 150 | 400 | do. | 11 | 165 |
| *Wilmington* . | 250 9 | 40 0 11/16 | 9 0 | 1,392 | T.S. Vert. Trip. | 15·08 | 1.894 | 17·08 | 100 | 300 | do. 4 Colts. | 10 | 165 |

## DISCUSSION.

Admiral Sir JOHN HOPKINS, G.C.B., LL.D. (Associate): My Lord, Ladies, and Gentlemen, your ever active Secretary has asked me to say a few words on this subject by way of starting a discussion. I did not come here to discuss the paper, otherwise I should have studied it more closely than I have done. It is an admirable up-to-date contribution, giving us all that has gone on lately in the American Navy in the way of construction, and we are very much indebted to Professor Biles for it, although, perhaps, it is not for me to say so. Probably it will not be out of place if I make a few obvious comparisons from an officer's point of view as to the differences between the American and our own battleships, and where the difference lies. They have in their latest battleships, I think, a fair equivalent to that of our latest battleships, only a little more so. In size they have cribbed 1,000 tons, and I think they are right. I believe Sir William White wanted his latest ships 3,000 or 4,000 tons larger, but there was such a howl against it that they did not give it to him; and now I believe he has fallen back on the 15,000 tons, which is our normal size, I think for many good reasons. If we increase the size of one or two ships, we destroy the chance of moving together as one, which we should be able to do if the vessels were all of the same size, and this is very important in naval

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tactics. If battleships can turn on the same circle, it will very much facilitate the manœuvring; for, in time of action, a Commander-in-Chief would be aware, what his fleet was doing, because he would know that they were all turning alike. I think that is a good reason for keeping these ships, if we can do so, all about the same size. The Americans apparently have not confined themselves to any particular type in regard to dimensions, but they keep on improving. I am sure you will agree with me that should be our aim, and we ought to be improving in a great many things: speed in our smaller vessels, in particular, and this applies especially to scouts and look-out vessels. I see the Americans have frequently run up to 19 knots in their battleships, and they have always gone one better in each succeeding type, or practically so. If you look at their distribution of armour, you will find it is quite up to date, as far as our naval ideas go. Take the *New Jersey*, for instance. No wonder I am told, when I walk round the dockyards, and say, "What a fine ship," the reply is, "Oh, yes, Admiral, she is, but she has soft ends." You will not see any soft ends about the *New Jersey*, because she has got a belt all round her. If you look at this American design, you will see that they have armour all over armament and vitals —they have armour on what we call the main deck, and that is the place where you want armour. You require it there, because you have the base of the funnels there, and several other things, which, if hit, would be very inconvenient indeed. We have plastered armour over the main deck of our latest ships and not on the side; but, in my opinion, we want to keep the shells out, and not to let them in, and then prevent the pieces going below. The way to do that is to carry your 6 in. armour higher up. The Americans have wisely introduced the 8 in. gun, and I dare say that we shall do so in the later designs. That gun is a very valuable gun, because, as Professor Biles says in his paper, a shell from it will penetrate, or ought to penetrate, 6 in. armour. We have all the armour which is now protecting a good deal of the auxiliary armament penetrable by the 8 in. gun. I see that two pairs of those 8 in. guns are superposed. I cannot hold with that, because I think each captain of a gun will want to get his own shot off in the direction of the enemy. The difficulty will be, who is to control the four guns? One man with us controls the pair. If the same man controls the four, both at the top and at the bottom, I think it will lead occasionally to little differences of opinion between the gunner on the top and the gunner down below. I should say that is one grave objection. Now, we come to a thing which the Americans obtain and we do not, small draught of water. That is very valuable indeed to us who want to send vessels through the Canal. When I was in the Mediterranean some time ago, the *Victorious* went through the Canal. That is a case in point. She had to get all her coal stores, ammunition, provisions, &c., out in the end. But prior to this, whilst drawing 28 ft. of water, she ran on a 27 ft. patch at the entrance of Port Said, and there she stuck. Through the mercy of Providence she was eventually got off. That is the result of having a heavy draught of water, which the Americans in these vessels have not; though what they actually draw at sea we do not know, as sometimes, in our own case, we are told that we have 26 ft. draught of water in our ships, but when we get them to sea we find it is 28 ft. That, however, is by the way. Another point is that they still stick to the two screws, which we have always adhered to. No doubt Sir William White, who is so clever a designer, has very good reasons for that in our case, but it has always struck me, seeing that some nations, the French, the Germans, and others, have in many instances fitted three screws, whether it would not be well that we should have a ship with three screws for a trial. The captains and the admirals do not want it, because it means that they would have another force to control from the bridge. Still, if three screws give an advantage in smallness of machinery, and also lessen the chances of a complete breakdown, and if, as some contend, there is economy with the triple screws in steaming at the slower speeds, then the

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three screws seem worth trying. Now I come to the boilers, though perhaps I ought not to talk about boilers, because I wrote a letter about them a day or two ago; but it is interesting to note that we do not see the Belleville boiler in these ships, and that seems odd. If they leave the Belleville boiler out, I have no doubt it is for some very good reason, because the Americans are very clever. But what I urge for our Navy is this: that we want the very best thing of its kind, whether engine, boiler, gun, armour, &c. But there is something we do not want, and that is for any novelty to be pitchforked wholesale into our ships, before we have found out whether it is the best of its kind. When the best type is found out, we ought to have it, and nobody will welcome it more cordially than naval officers. The docking keel is more a question for the naval architect, but it seems a very good idea to me. Perhaps the Americans were led to that by the misfortune that occurred to one of their ships in Southampton Docks some time ago. In structural strength I am led to believe that our ships will stand all ordinary docking strains. That was not so with the American vessel apparently, though it is generally supposed there was some neglect. Then, as to torpedo tubes, in the latest battleships they have two, and I think everybody will imagine that is enough. I have taken the battleship because that is sufficient to talk about. I daresay somebody else, if there is anything else to discuss about the cruisers, will kindly give us a few words about them. I am sorry to have taken up your time so long.

Vice-Admiral C. C. P. FitzGerald (Associate): Professor Biles did not tell us whether they had submerged torpedoes.

Professor Biles: There are four; you will find that in the paper.

Admiral FitzGerald: They have left out the above-water one.

Professor Biles: Yes.

Admiral FitzGerald: The only point I want to mention is as to those two superimposed turrets. When I was at San Francisco about this time last year an American officer kindly took me over one of the ships that was building there. I was rather surprised to hear Professor Biles say that the last idea was to have superimposed turrets, because I learnt from the American naval officers that they were all dead against them. They said they could not make anything of the shooting. The summing-up in Professor Biles's paper seems to indicate otherwise. It is a little inconclusive, but still, he says, the last design was to have the 8 in. guns on the top of the 12 in. I met a few naval officers at San Francisco, and they let me go over the Union Ironworks to see what was going on there, and they were all against the two-decker turrets. Also with regard to the 8 in. turrets not superimposed —the two forward ones—as Professor Biles has mentioned, when firing ahead, the men cannot work in the forward 12 in. turret, so that these 8 in. guns may perhaps be too heavily paid for. If the Americans can put in 8 in. guns, in addition to a similar armament to that of our *Majestic* class, then one is inclined to ask: What is left out? because something must be. What have they left out in order to get these features in? One cannot float a ton of mild steel with less than the displacement of a ton of salt water; so there must be something left out in these American ships. I should like to know what it is. There is one other point about the small vessels—these torpedo boats. They appear to have a vessel of 144 tons which goes 30 knots with reciprocating engines. If they have really done that they have beaten everything we have done in this country, and Mr. Yarrow and Mr. Thornycroft will have to sit up.

Professor J. H. Biles (Member of Council): My Lord and Gentlemen, there is not very much to which I have to reply. Admiral FitzGerald asked what was left out in these ships. It is

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impossible to say what is left out in the American ships as compared with British ships, because there are not any similar documents published in England, such as the American Navy Authorities publish, to give us information in detail as to what is in the British ships. If we had that information officially given, I should perhaps be able to say what has been left out in the American ships. If Admiral FitzGerald wants that question answered, he should take the information which is given so fully by the American Naval Constructors, and reproduced in this paper, and go to the Admiralty, where all the information as to what is in the British ships can be got, and he will soon find out what is left out of the American ships. I think, my Lord, there are no further questions to which I have to reply. I am not quite sure that Admiral FitzGerald has appreciated that the tonnage of these American ships is not necessarily the same as our ships, especially if some of the later American ships are compared with some of our latest British ships. But the Americans seem to be able to get more tonnage into a given sized ship than, in some cases, our Admiralty have been disposed to do. In other words, their ships are fuller than ours.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): I am sure we are all very much obliged to Professor Biles for the interesting paper he has just read.

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# A DESIGN FOR A FAST SCOUT.

By Vice-Admiral C. C. P. FitzGerald, Associate.

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I beg to lay before the meeting a design for a scout of high speed and good sea-going qualities.

I am responsible for the general idea of the design, but, not being myself a ship designer, I have been obliged to ask for professional assistance, and Mr. Philip Watts, of Elswick, has most kindly taken all the trouble to work out the design for me, and to carry out my views, so far as that was possible on a given tonnage; though, of course, we naval officers always want more than we can get, and more than the laws of hydrostatics will allow.

I have also gratefully to acknowledge the assistance of Admiral Sir John Hopkins, whose great experience and sound practical views on all naval subjects are universally admitted.

The proposed vessel is to be of 3,800 tons, 400 ft. long between the perpendiculars, 44 ft. beam; her draught will be 14 ft. with her normal coal supply on board; there will be two screws, and the collective horse-power will be 17,000. The armament is to be six 4 in. guns and a dozen machine guns. There will be a protective deck 2 in. thick on the slope, and 1 in. on the flat. There will be no double bottom through the boiler compartments, but there will be elsewhere. The 4 in. guns will be protected with 4 in. shields, except the foremost and after guns, which will be in 4 in. gunhouses, and the conning tower will have the same thickness of armour. The speed is to be 25 knots; but the whole idea of the design has been, not so much to procure a very high measured-mile speed for a few hours, as to assure a good, continuous ocean speed, as long as the coal lasts, and this is to be 23 knots. The normal coal supply is 500 tons, but the bunker capacity will be for 1,200 tons, though I need scarcely say that the vessel will not go 25 knots with her bunkers full.

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The cost, without armament, will be about £250,000; and the armament will cost about £20,000; including 200 rounds per gun for the 4 in. guns, and 500 for machine guns.

The radius of action with full coal stowage will be approximately as follows:—at full speed, 1,500 miles; at 23 knots, 2,000 miles; at 18 knots, 3,000 miles; and at 10 to 12 knots, 8,500 miles.

The vessel could carry either one 56 ft. vidette boat, or two 48 ft. boats, in addition to the ordinary rowing boats.

The original idea was to have a smaller vessel with very quick-running engines; but, taking into consideration the liability of such engines to give trouble when working continuously at sea, it was decided to have less quick-running engines, and, consequently, a larger ship. The revolutions will be under 200 per minute at 23 knots. The boilers will, of course, be tubulous; probably of the Yarrow type. The ventilation will be artificial, and there will be no cowls.

Turbine engines were contemplated; but, after full consideration, reciprocating engines have been decided upon.

It seems necessary that I should show cause for proposing a design for a ship of war of this tonnage, with such a very insignificant armament.

In the first place, then, I would beg to point out that she is not intended to be a "ship of war" in the ordinary sense of the words; that is to say, she is not intended to be a fighting ship, and thus the proposed armament must be regarded as altogether of secondary, or rather of tertiary, importance. The two first qualities aimed at are continuous high speed, and good sea-going qualities in all weathers. It will, therefore, prevent confusion, if we call her a despatch vessel, or a scout.

She may not make much show on paper, though that is a matter of small importance, if it can be proved that such a vessel, or rather a considerable number of such vessels, would be of the most vital importance to enable us to carry out our avowed strategy in the event of war with a maritime Power; and notwithstanding that the want of such scouts may not become apparent until war is actually upon us, or, in other words, that there is no use for them in peace time. And in this connection I may remind you that, at the time of the Russo-Turkish war in 1878-79, when our squadron passed through the Dardanelles, and it seemed probable that we should be involved in hostilities, the want of fast scouts became so urgent that the Government were obliged to buy up hastily a number of Liverpool tugs. Our own small craft of the day were so deficient in speed and coal endurance that they were practically

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useless for the duties of scouts and despatch vessels, notwithstanding that they could sail.

It has often been suggested that we ought to keep ourselves ready for war at short notice, so as not to find ourselves in the position of having to extemporise any important class of vessels as auxiliaries after war is declared, or becomes imminent; and, with regard to scouts, there do not appear to be any vessels in existence which would meet our requirements. The Transatlantic liners are too large. The Holyhead boats and the Channel Island packets are fast vessels and good sea boats, but their boilers and engines are unprotected, and their coal supply altogether insufficient to enable them to act as scouts; whilst our old friends, the Liverpool tugs, are scarcely up to present-day requirements with regard to speed.

It is generally understood that, in the event of war with a maritime Power, it would not be possible to institute a close blockade of the enemy's ports in the same way that we blockaded during former naval wars; but that it would, nevertheless, be absolutely necessary that we should be able to watch those ports in such a manner that the exit of hostile ships in any considerable numbers could not take place without the admirals in command of our battle squadrons being promptly informed of the fact. But, although this is the leading feature of our avowed strategy, it is difficult to see how it is to be performed, or what warships we have capable of doing it. To begin with our first-class cruisers: they are too valuable to be exposed to the great risk of watching off an enemy's port, which, probably, would contain numerous torpedo boats, and, perhaps, some submarines; secondly, there are not enough of them; and, thirdly, they would have quite enough to do in protecting our trade routes from the depredations of hostile cruisers, which are certain to escape, in spite of the closest watching. Next, take our second and third class cruisers, with nominal speeds of about 19 and 19½ knots; some of the smaller ones (the "P" class) having a speed of 20 knots in smooth water: it does not seem likely that they will be permitted by an enterprising enemy to keep watch off a port which will probably contain several powerful armoured cruisers of large size, and with nominal speeds of 21 knots.

These latter vessels would appear on the scene some fine morning at daylight, with a good head of steam ready, and sink or capture our watchers, as the latter would not have sufficient speed to escape, even under the best conditions.

The fact that our second-class cruisers have not improved in speed during the last ten years was forcibly pointed out by Admiral Sir John Hopkins a short time ago, in a very interesting and comprehensive paper which he read at the Royal United Service Institution. The increase in size in the more recently built vessels of this class, and the saving of weight effected by the use of water-tube boilers, have been devoted to giving them a more powerful armament, but without affording them any material advance



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in speed (half a knot nominal,* I think it is). It is not for me to criticise this policy. I merely point out that these later vessels do not appear to be any better suited to perform the duties of scouts than their older sisters; in fact, for some reasons, rather less so, as they cost about £50,000 more money.

Our torpedo gunboats have not sufficient speed, or sea-going qualities, to enable them to act as scouts to our battle squadrons; and, lastly, our destroyers have neither the size, sea-keeping qualities, nor the coal endurance to fit them for the duties; though they are doubtless excellent little vessels for the purpose for which they were intended.

Recent events in South Africa have illustrated the extreme importance of efficient scouting in land warfare; and, if we may reason by analogy, there seems to be cause for believing that scouting will be of at least equal importance in the event of a great maritime war.

I have thought it desirable to enter at some length into my reasons for proposing a vessel of this tonnage, speed, and horse-power with such a very light armament, though I am ready to admit that such vessels are not wanted in peace time.

I now submit the design for your criticisms. (See Plate.)

## DISCUSSION.

Commander Richardson Clover (Naval Attaché to the American Embassy, Associate): My Lord and Gentlemen, during the short experience that we had in our war two years ago or more with Spain, we found great need for scouts, particularly in the early part of the war, and we employed ocean liners, such as the *St. Louis* and the *St. Paul*, for that purpose, and found them very efficient. The result of our experience has not been that it is desirable to build vessels especially for this purpose. A vessel in order to be an efficient scout must have good size, in order to maintain its speed at sea at all times. With a vessel of only 3,000 or 4,000 tons high speed cannot be maintained in a heavy sea, particularly when making a long passage. I therefore believe that the ocean liner, with its availability, its assurance of maintaining high speed, and its large coal capacity, is better fitted for a scout. I am speaking now of the long-distance scout, not of the work that can be done by torpedo destroyers, which can fulfil the work of a small vessel, but it was our experience that we had to send these larger vessels to cruise long distances. Not one ship only, but a number were required, so that, if we had to build special vessels for scouting purposes, we should have to build a fleet of them. Our conclusion was that dependence could best be placed upon ocean liners, or fast auxiliary cruisers; and for close scouting we found that our torpedo boats and torpedo cruisers were efficient.

* The latest official intimation says 21 knots; that is to say, 1 knot better speed than our present second and third class cruisers.

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Admiral Sir JOHN HOPKINS, G.C.B. (Associate): My Lord and Gentlemen, I would much sooner have had Sir William White upon his legs before I got on mine, for he might make objections to this design, because it was not got out at the Admiralty. However, as I could not get him to speak first, he will be in a position to pour a broadside into me when I sit down. What I want to say about the scout is this. I have had a little experience of the want of scouts, not absolutely in war time, but in making preparations for war, and in making these preparations one has, *inter alia*, certain places to watch very closely. With all due deference to my American friend, I prefer for this duty our own vessels, and not Transatlantic steamers of the White Star Line, or any other line, of 10,000 or 12,000 tons, for these huge vessels would not so efficiently do what would be required from them. The work I wanted them to do in the Mediterranean was to watch certain strategical positions, say the Straits of Messina, the Straits of Bonifacio, the entrance to the Dardanelles, the Malta channel, and various other places. The dilemma in which I found myself was this. We had plenty of second-class cruisers which steamed 20 knots; a measured mile 20 knots, not 20 knots sea speed. What I felt was that, if they got into a position where they might be caught by a hostile cruiser of 21 or 22 knots, I should not get the information I wanted, and, therefore, I should not be able to make my arrangements. A good deal of what we do is founded on history, and a fine page of history it is. Look at our greatest hero, Nelson. What did he say? "The want of frigates would be engraven on his heart." He did not want them to fight other frigates. He wanted them as scouts; he wanted the enemy watched, and he wanted to get information as to the whereabouts of the enemy, and that information he wanted brought quickly by means of his frigates. If his scouts were unfortunately caught by the enemy's vessels, he would not get the information needed. The same with our scouts. If they were run down by anybody else's bigger and faster ships, we should not get the information. If I required to watch various points, and I had nothing very fast to send on this duty, there would be the danger of the vessel being taken, and then I should not get the information. That is the position in which I stood as Commander-in-Chief in the Mediterranean. Therefore, I shall keep on, in season and out, urging on the authorities and on the constructive department to build cruisers which can go something better than this wretched 20 knots. Why do I say wretched? Ten or twelve years ago Sir William White produced a class of vessels far ahead of anything of that day, and they went 20 knots. Now, apparently, either his genius is fettered, or the Admiralty does not want fast small cruisers; which it is I do not know, but the Service wants these ships. I have no doubt that if the Admiralty ask for a scout able to satisfy the requirements which I have mentioned, Sir William White will be able to design one. We see all round us foreigners building their 23 or 24 knot vessels. In a paper read at the United Service Institution the other day I dotted down a few facts which I will give you. I will first read you what Captain Mahan says about this, and how we need to sacrifice other qualities to get speed. The words are these:—"In the primary requirement every other feature should be subordinated. In cruisers rapidity of action is their primary object. Defensive strength does not conduce to rapidity of movement, nor does offensive power, they must, therefore, be very strictly subordinated." So you see he sacrifices for speed defence in armour and offence in the way of guns. "They must, therefore, be very strictly subordinated." That is exactly what has taken place in Admiral FitzGerald's scout. Everything is subordinated to the highest speed, and, in my opinion, rightly so, unless some of the naval architects can produce a vessel of this class, which has also a good armament, and then we have nothing further to wish. There is a clever officer, who has written two prize essays for the United Service Institution Medal, Commander Ball, and what does he ask for? and he does not think he is asking for the

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impossible. He asks for 23 or 24 knots in our first-class cruisers, and that speed we have reached, I am happy to say, though rather late. He asked for 22 knots in our second, and 21 in our third class cruisers, and he is a man of moderate views with regard to speed in small classes. We look at what foreigners are doing. We know that the *Guichen* went 23½ knots some time ago on a three hours' trial. People said that it was only a trial speed. Ours are very much trial speeds, too. There is the *Jurien de la Gravière*, which has a speed of 23 knots, and carries eight 6·4 guns, and the *Askold*, a 6,000 ton Russian vessel, with twelve 6-in. guns of 45 calibre, to go 23 knots, and out of which they hope to get 24. In the face of all this, are we to stand still and get no higher speeds? I put it to you, as an assemblage of naval architects, should this thing be? All I am attempting to do is, in and out of season, to go in for speed in certain classes. I said some time ago that our battleships should go 19 knots: so they can now. I know Sir William White could do it, if he liked; but probably he had a circumscribed area suggested to him outside which he was not to go. With regard to this vessel, in my opinion she furnishes much that we want in the shape of a scout. Of course, we do not know but that this vessel may not be beaten by the Admiralty designers, and we hope it may be so; but, as a sailor, I say that this is the sort of craft a commander-in-chief wants put at his disposal to watch, to obtain information, and to bring it in quickly. The American officer who spoke said his Government had quite enough to build without building scouts; but I do not think that ought to be said of our Government. If necessary, our Government should build them. It is not a question of whether they want to, or care to; the point is, if vessels of this class are necessary, they should be produced.

Captain H. J. May, R.N., C.B. (Visitor): My Lord and Gentlemen, after all we have heard from Sir John Hopkins, I speak with a good deal of diffidence; but I want to raise one or two questions about this ship, so as to get a clearer naval officer's view as to what she is and what she is not. Perhaps I may say what I understand her to be. I understand her to be a ship mainly intended for running away, with almost a certainty of escaping, from a second-class cruiser. But it is doubtful whether she would be able to escape from a first-class cruiser, looking at her small size, and having regard to the fact, which always arises in a small craft, that there is great difficulty in keeping speed at sea. Therefore, if she is watching some port or other position, she can be driven off her station by a second-class cruiser, because her armament is so small. She would scarcely be driven off her station by a third-class cruiser, and she could drive away, and possibly catch, a destroyer. If I am wrong, I hope the lecturer will presently correct me. Now, we will suppose we want ten of these craft. Then we should have to go to the House of Commons for the men and the money, and we should have to say: "We want ten scouts." They will cost some 2¾ millions, and, besides this, we shall want the crews. We will put the crew at 300 (no estimate has been given on this point, and, if I am wrong, I shall be corrected), so we shall want 3,000 more men. If you look through the Naval estimates, and take all the money that is devoted to *personnel*, including victuals, &c., and divide that by the number of efficients actually available at the present time, you will find that about £100 per head is required for each person actually serving. So that £300,000 must be added to the estimates year by year to pay the crews, and all this for ships which the lecturer said were not fighting ships. Or, if they are to some extent fighting ships, they can only fight a third-class cruiser or overpower a destroyer. And now to go back to history. Nelson wanted frigates. What was the frigate that Nelson needed? What did Nelson know of a frigate? If we look at the history of the British Navy, we shall see that British frigates nearly always beat French frigates. Nelson then knew well that the

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vessel he required would beat a French frigate, if she were required to do so. He did not ask for sloops or lightly armed vessels. Moreover, we know that when the Americans produced frigates which the British frigates could not beat, we were in a great state of mind, and had to give an order that no British frigate should go out singly, but that they should go out in pairs. Therefore, it appears to me that in producing this second-class cruiser, if I may so call her, which is to run away from a hostile second-class cruiser, we are not producing a Nelsonian frigate. When it comes to a question of scouting, so far as my study of the manœuvres and other recent operations has gone, a scout comes up with the enemy's fleet, or approaches the enemy's port, and the first thing she sees on the horizon is a hostile cruiser. She closes in until she discovers whether the other cruiser is bigger than herself. If she is bigger and is making rapidly towards her she must make off to avoid capture. It has often happened both on paper and in practice that the scout never sees anything else but the cruiser that chases her. The cruiser chases her twenty, or thirty, or forty miles, and if she is fortunate enough to escape and go back to the admiral, she reports that some hours ago, some hundreds of miles away, she was chased by a cruiser and escaped. It is not very satisfactory for the admiral, and in manœuvres, at any rate, our admirals have had a good deal of this kind of information, which has been of very little service to them. Therefore, I do not honestly see how we can go to the country and say we want these expensive ships, because I do not see where they fit in.

Sir William White, K.C.B., LL.D., Sc.D., F.R.S. (Vice-President): My Lord and Gentlemen, my friend, Sir John Hopkins, has assumed that I should adversely criticise this design. He gave as his reason that I might be influenced by the consideration that it was not prepared in the Admiralty. Now I have no idea of criticising the design. There are certain points about it which are of great interest to all who are concerned with warship construction. I may, however, mention one circumstance which is of interest. About three years ago in the French Navy estimates there appeared a proposal to lay down two vessels (*croiseurs d'estafettes*) very closely approaching in dimensions and in general characteristics the vessel which Admiral FitzGerald has suggested. Those vessels were described in some detail. A grant was proposed to be taken for beginning them. They have not been begun, and the grant has not appeared in more recent estimates. There was a considerable discussion at that time in the French Service papers as to the value for scouting purposes of lightly armed swift vessels of moderate size, but of very large cost in proportion to their size. Without going over the details of that discussion, I may say that the balance of opinion seemed to be distinctly on the side which Captain May has just supported. Of course, it goes without saying that if size is to be kept down and speed made very high there must be sacrifices. Admiral FitzGerald has put that quite clearly. I think it is impossible to make a fairer statement of the case than he has made. There are, however, certain points in this solution of the problem which are of importance. An engine running 200 revolutions per minute for continuous steaming may be, and is, by comparison with destroyer engines, "slow-running engines"; but these are absolutely quick-running engines, and, compared with the machinery of swift sea-going steamships of large power, they would be very quick-running. Mr. Watts, of course, has gone into this matter with full knowledge, and I think he will bear me out in what I am going to say. The power produced with this type of boiler over long periods of continuous steaming must drop very considerably as compared with the maximum performance on a short trial with everything at its best. That is a necessity of the case. On the basis of experience with ships fitted with similar boilers, the drop in power for long periods may be 40 to 50 per cent. from the maximum power on a three hours' trial. In view of this experience the estimate of 23 knots for continuous steaming at sea seems to me somewhat sanguine.

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But it is obvious that between 23 knots continuous speed at sea and anything that is to be found in existing vessels of this size there is a large margin, and the proposed scouts would have relatively high speed. I do not propose, my Lord, to say a word in reply to the very pointed and spirited criticism of Sir John Hopkins on Admiralty policy. That would be out of place on my part, but I thank him personally for his very kind allusions to myself.

Mr. Philip Watts, F.R.S. (Vice-President): With reference to the remarks made by Sir William White, the normal displacement of the ship is made up approximately as follows:—Hull and fittings, 1,840 tons; propelling machinery, 865 tons; protective material, 330 tons; armament, 130 tons; equipment, 150 tons; coal (as Admiral FitzGerald has told us), 500 tons. The 865 tons would be sufficient to provide two sets of four-cylinder engines with a 2 ft. 6 in. stroke, capable of developing at 230 revolutions about 17,500 I.H.P., steam being supplied from sixteen Yarrow boilers of ample dimensions. With three-fourths of the boilers in use, rather under 200 revolutions could be maintained, as Admiral FitzGerald stated, which would give a speed exceeding 23 knots. There is no doubt about that, I think.

Sir William White: That is 75 per cent. of the maximum performance of the boilers.

Mr. Watts: Yes.

Sir William White: That is the point I meant.

Mr. Watts: That would leave a quarter of the boilers always in course of being cleaned. It would be, as Sir William White has indicated, a very good performance.

Sir William White: A very high performance, if continuous.

Mr. Watts: We should get 48,000 sq. ft. of heating surface for the full power (17,500 I.H.P.), which gives $2\frac{3}{4}$ sq. ft. per indicated horse-power; and with three-fourths of the boilers in use we could get 12,000 I.H.P. with 3 sq. ft. of heating surface per indicated horse-power, which would probably more than suffice for the 23 knots. I should hope to obtain the 23 knots with 11,500 I.H.P. I am thus assuming that about 66 per cent. of the full power would be maintained at sea, which is only 6 per cent. more than the upper figure named by Sir William; so that there is not very much between us as regards the power. Sir William referred to the number of revolutions as very high. No doubt, 200 revolutions for continuous steaming is high, but experience has, I think, shown that this could be maintained with well-made and properly balanced engines, such as I have described, and such as have been fitted by my firm in cruisers for the past fifteen years. On these points I have asked the opinion of engineers who have had experience with machinery and boilers such as are here contemplated. Mr. Robert Humphrys has written me within the last few days that he is firmly of opinion that the performance could be confidently looked for, and his firm is prepared to supply such machinery, and undertake that it shall fulfil the required conditions as to power and continuous steaming. Mr. Marshall, of Hawthorn, Leslie & Co., Ltd., is also of opinion that with this machinery the performance contemplated would be realised. Mr. Yarrow, writing with reference to the boilers, says, "I certainly think you ought to get two-thirds of the full power for continuous steaming, considering that it works out at over 4 ft. of heating surface to the horse-power." I do not know that I have any other remark to make. This is not my design, but I hold myself responsible for the speed. I feel confident that the vessel could steam at the speed named in ordinary weather as long as her coal lasted. The quick-running machinery was necessary to obtain this performance on the displace-

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ment given. A bigger ship would, no doubt, have certain advantages, but Admiral FitzGerald has attached importance to keeping the vessel as small as possible. I should have preferred a heavier armament in the ship to give her a chance against cruisers of about the same size. Two 6 in. guns in gunhouses, and torpedo tubes into the bargain, could be added at the comparatively small cost of, say, £20,000. This would increase the displacement by about 200 tons. By adding another 200 tons you could replace all the 4 in. guns by 6 in. guns at a further cost of, say, another £20,000. Her displacement would then only be about 4,250 tons, and she would be a well-armed cruiser, and, if not intended to seek battle, could defend herself against similarly armed vessels when unable to avoid attack.

Professor J. H. Biles (Member of Council): My Lord and Gentlemen, I should like to say a few words on this design; not so much as to the details, as upon the question of the advisability of building such ships. Commander Clover has told us the practical experience of the American war. Some people are very fond of practical experience, and amongst these I think the British naval officer is as prominent as anyone. Therefore, I think I am justified in saying what I am about to say on the desirability of relying to some extent upon our high-speed passenger steamers as scouts in time of war. Some few years ago I had the honour of reading a paper before this Institution at Southampton upon the possibility of so designing mail steamers that they could be adapted in war time to taking up, to some extent, the duty of fighting, and certainly of taking upon themselves the duty of scouting. The design we have before us does not compare in offensive power with the mail steamers as they were armed in the American war. This vessel is not so powerfully armed as our own vessels could quite readily and quickly be in the event of war. Therefore, as a fighting machine, this vessel is certainly directly comparable with a mail steamer. Now as to protection. The methods suggested in my paper of adopting the design of mail steamers, so that in time of war they could very readily have protective armour put upon them, were, I think, feasible. There was no very strong objection urged against them, and they could, therefore, I conclude, be brought into as good a defensive condition as this design. Admiral FitzGerald asks for a vessel that costs £250,000. Now a good mail steamer, certainly one bigger than this scout, could be built with that money, and it might include within itself, as well as this vessel could, the possibilities of defensive and offensive warfare; perhaps not quite the same trial-trip speed, but certainly the same average sea-going speed. A vessel could be built at very little more expense, or perhaps at the same cost, which would in time of peace be giving some return for the money spent, and which would in time of war be able to do the ordinary operations of scouting. Captain May has said, as I understood his argument, that scouts are practically useless, because they are chased off by other cruisers, and, therefore, the Admiral gets no information but that the scout has failed in its duty. Of course naval officers will settle between themselves as to whether there is a necessity for scouts or not. I should suppose there is, and, if so, I am disposed to think we ought to attach considerable importance to the view Captain Clover has put forward, that mail steamers, if properly designed in time of peace for such purposes, will make as effective scouts as we can get, and that we can then devote our resources to the building of warships which will be able to fight, and need not run away.

Vice-Admiral C. C. P. FitzGerald (Associate): My Lord, I am very much obliged to the gentlemen who have cut up this ship and fired into her and sunk her. The matter seems to me to resolve itself into the question of whether such a vessel is wanted, or is not wanted. I wish I had some more of my friends here who are now commanding squadrons. I have received letters, which,

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unfortunately, are private, and which I cannot read, saying they think this vessel is needed. Admiral Hopkins' views are, as all know, of the highest value, and he has told you his opinion. I will now take the objections which have been made in detail. Commander Clover does not think such vessels are necessary. I am very much obliged to him for giving us his practical experience; but I would venture to point out that the conditions under which the *St. Louis* and other similar ships were used in the Spanish-American war have very little connection with what we should want in England in case of war with a maritime Power, as we are so fond of calling our nearest neighbour. These merchant steamers did their work; but, no doubt, during the Spanish-American war they were, at best, makeshifts. Are we to be satisfied with makeshifts? In the same connection Professor Biles backed up the use of these makeshifts for us in case of a maritime war; but it must be remembered that they burn an enormous amount of coal, they are huge vessels fitted up for passengers, and they have not the speed of my proposed scout. I always understood that these large and fast merchant vessels would be wanted to feed the country, there are no other vessels which could get across the Atlantic quick enough with corn, if we were involved in a great war. Our numerous 10 knot cargo steamers would be useless, and every one of these large and fast ships would be wanted to do their own duty as merchant steamers. They might be armed, of course; and they might have naval officers on board them, and fly the man-of-war ensign, so as to be allowed to fight without being sunk as pirates, but they would certainly be wanted to bring provisions into the country. They would not be efficient as scouts. They have been so used at manœuvres, the vessel that was lost, the *Oregon*, was so used; and they compared well with the vessels we had in those days, because they could go, and the others could not, and they could keep it up at sea. In the American Navy, in the same way, their small cruisers were not up to the work. One class of vessel could carry on the duty, the other could not, and they had to use the one that could. It was not the best for the business, but it was the best they had ready. Captain May cut my scout up severely, and said she was not wanted at all. But, when he said she was only intended to run away, I venture to think he rather strengthened my case. He described to us somewhat humorously the duties of a scout. To come up, and see a big cruiser, and to say to himself, "She is bigger than I am," and then run away, and tell the Admiral. That you may say with regard to all scouting. In land scouting you send a man out on his dangerous errand, if you want to locate the enemy, but you cannot always expect the scout you send out to see the whole state of affairs within the enemy's camp, nor at sea to see what his battleships are doing, but he gets in touch with the enemy, and probably obtains useful information. Your scouts in land warfare do not fight. I mentioned that point in my paper, because I think the cases are somewhat analogous. We have seen in South Africa the terrible results of the want of proper scouting and locating the enemy, and the same thing would occur no doubt in a naval war. It seems to me that the value of good scouting is beyond all question. I need scarcely say that I am exceedingly obliged to Mr. Watts for his services. He says he is not responsible for the armament, and that, of course, is true. We began at 2,500 tons and had a poop and forecastle, but they gradually grew and met amidships, and then we got up to 3,600 and 3,800 tons, and now he wants to go to 4,300 tons and to carry 6 in. guns. But I would point out that this game of increase is unlimited. The question is one of cost, and whether it is not better to have a vessel deliberately intended for running away and bringing information, rather than to build fewer of a larger size intended to stop and fight. I say that a scout should not stop and fight. If he stops and fights, he is not doing his duty, because what he is required to do is to come back with the news. My scout is not supposed to contend with a second-class cruiser of the enemy, and she does not perform the *rôle* for which she is built, if she

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does so. Sir William White told us that somewhat similar vessels were designed by the French; but they were never built, because the French did not consider that they wanted them.

Sir WILLIAM WHITE: That is so.

Admiral FITZGERALD: It is quite possible that the French may not require a scout of this sort, but that we do. I understand our strategists differ totally from each other. It is, therefore, quite possible the French do not want them, and are right in not building them, but that does not seem to be a valid argument that we do not want them. Our proposed strategy is of a different nature. We think we are going to watch off the enemy's ports. Our whole policy is based on the idea that we are to be stronger at sea and to watch the enemy's ports closely; and hence the need for scouts intended to come back and give the Admiral information as to what they have discovered. They might of course be driven off by a crowd of cruisers, just as in land warfare a cloud of cavalry is used to mask the movement of the enemy's main body; but, if we had not scouts, we should not have the ghost of a chance of finding out what our enemy were doing. How our present second-class cruisers are going to do this I cannot see. The sole object of my scout is to come away and bring whatever information she can obtain. Now, with regard to speed. Sir William White thought it was a very sanguine estimate that this vessel should be able to keep up 23 knots continuous speed. That is a technical question into which I do not feel competent to enter. I am entirely in the hands of the designer on this point. It is said by Sir William White that 23 knots sea speed is better than has ever been done before; but Mr. Watts has several times done better than anything that was ever done before, and I do not see any reason why he should not do so again, and I therefore pin my faith on him. There is another matter with regard to the use of merchant steamers as scouts, and that is that they have exposed engines. Professor Biles told us that merchant steamers could be built —he did not say that they existed—in such a way as to be able to take the place of scouts in time of war, and perform their own mercantile duties in time of peace. I do not doubt it; but they are not built, we have not got them. Even if we had, and they met anything that fired at them, they could not compare with my vessel, because the latter has a considerable amount of protection for her engines. The merchant steamers may carry a more powerful armament; but, if you can easily destroy the engines, she is utterly disabled. The engines come right up to the upper deck. The engine-room of the *Campania*, for instance, which I came home in the other day, was as big as this hall, and went right to the upper deck. One shell bursting in it would probably disable the whole of the propelling machinery. These steamers also burn an enormous amount of coal, and that will be an extremely important matter in war. I think I have mentioned all the points raised. I have a comparison here with other ships—the German *Niobe* and the Russian *Novik*, and the Russian *Boyarin*. They are different vessels and much more powerfully armed, but they have not the speed. There are many other points which they have not got, and they cost a good deal more money. However, that point has not been mentioned, and I need not go into it. I am much obliged to you, gentlemen, for the attention you have given to my paper.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): I beg to move a vote of thanks to Admiral FitzGerald for his admirable paper. I think we have all listened to it with much interest, and also to the able speeches which have been made by the other gentlemen who have followed him. You have all your own opinions, I suppose, as to who carried away the honours, but I should say they were pretty equally divided.



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# AN INSTRUMENT FOR MEASURING THE ROLLING OF SHIPS.

By A. Mallock, Esq.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 27, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

At first sight it might appear that nothing would be more easy than to measure the angle through which a ship rolled, and that all that would be necessary would be to hang a pendulum anywhere in the ship, and to observe the angle which it made with a fixed line, which was vertical when the ship was upright. In reality, however, the matter is not nearly so simple. Let anyone take a fishing-rod, with only one or two feet of line out, a weight being at the end of the line. Let him wave the rod from side to side, and it will occasion no surprise to observe that the hanging line does not remain vertical during the process. The line and weight in this case exactly represent the pendulum on board a rolling ship (except in the special case where the rolling is executed in still water, and the point of support of the pendulum is at the centre of gravity of the ship). If the point ot support be anywhere above, or below, the centre of gravity, it must move sideways when the ship rolls, and as the only connection between the bob of the pendulum and the ship is its point of support, the bob can only be set in motion sideways by the line joining the point of support and the bob not remaining vertical; *i.e.*, by the horizontal component of the force transmitted along the pendulum from the support to the bob.

On the supposition that the angle of roll remains constant a very simple construction, which will be given below, enables the angle which the pendulum makes with the vertical at any part of the roll to be calculated; but when the rolling is irregular, as it generally is at sea, the departure from the vertical will be much greater; and to deduce the true value of the roll from ordinary pendulum observations in such conditions would involve lengthy calculations, and a knowledge of the length, height, and direction of the waves, obviously impracticable in ordinary sea life.

The only methods of measuring rolling which fulfil the practical requirements of sailors are such as give direct readings; but hitherto only one such method has been

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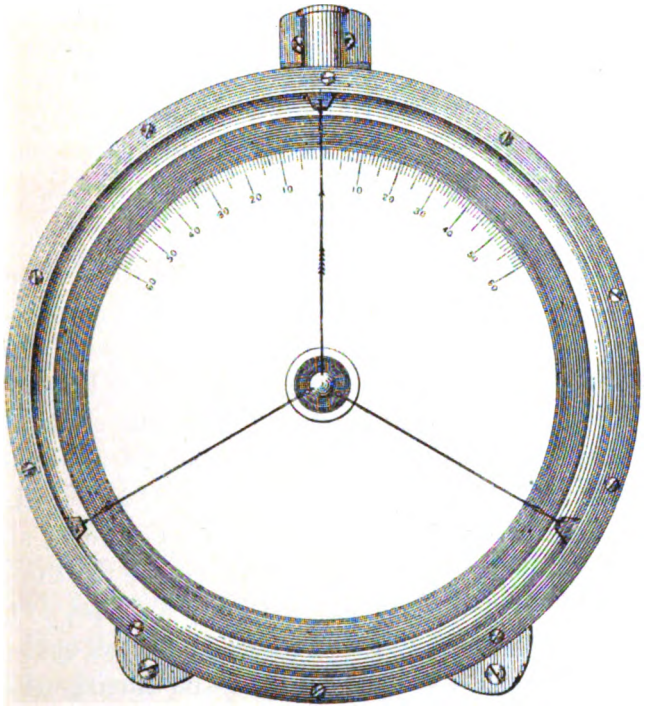
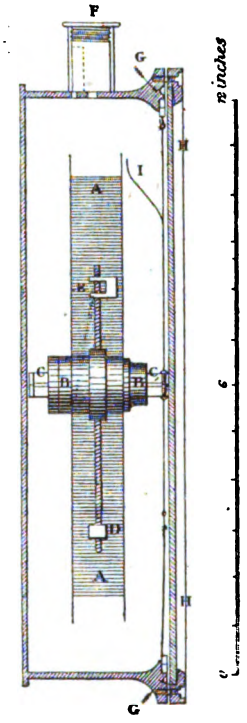
in use, namely, observations with battens. Upright battens, fixed near the ship's sides, are divided by a scale of tangents so as to read degrees when observed from a fixed point amidships, and the division on the scale, which is cut by the horizon when the angle of roll is a maximum, obviously gives directly the angle of roll if the reading is zero when the ship is upright. As regards accuracy, batten observations leave nothing to be desired (for such errors as are caused by the variations of the observer's height above the sea level are of too small an order to be considered in this connection), but they are always more or less troublesome, involving two observers, and cannot often be made in bad weather, nor, of course, at night.

FRONT ELEVATION-WITH BRACKET SUPPORT



SECTION-WITHOUT BRACKET




A A.—Wheel.
B.—Hollow boss.
C C.—Jewel bearings.
D E.—Counterpoises for stability, and zero adjustments.
F.—Filling hole and expansion box for fluid.
G.—India-rubber joint.
H.—Plate glass.
I.—Index pointer.

The Rolling Indicator here described may be placed in any part of the ship, and will give a very accurate measure of the rolls in all circumstances. The errors to which it is subject and their sources are treated below in sufficient detail to enable anyone interested in the matter to follow the steps by which the results are reached. The Indicator consists of a paddle-wheel supported on fine pivots, and enclosed in a short cylindrical box completely filled with fluid. The wheel is made of the same

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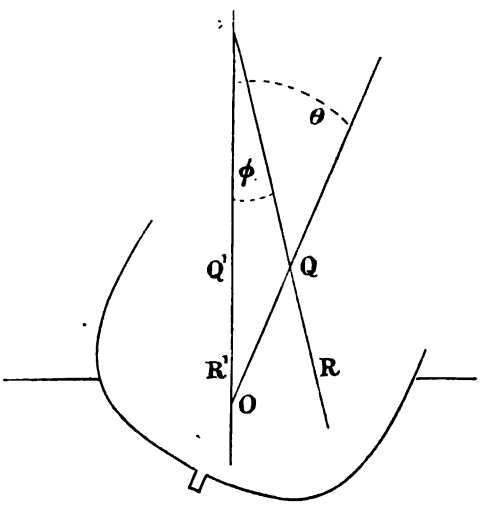
density as the fluid, so that the friction of the pivots is nearly eliminated. It is well known that in the case of a box completely filled with a perfect fluid, no motion given to the box can communicate any rotation to the fluid itself; and, though this is not true of a viscous fluid like water or any actual fluid, it is practically true of that part of such a fluid far removed from the sides, provided that motion is not sufficient to cause eddies. The paddle-wheel nowhere comes within an inch of the sides of the cylindrical box, and experiment has shown that this distance is sufficient.

Under these circumstances, whatever motion is given to the box, very little rotation is set up in the interior of the fluid, and the wheel, which is of the same density as the fluid, behaves as if it were part of it. In fact, the function of the wheel is merely to indicate what the inner parts of the fluid are doing. When the ship rolls the box revolves, but leaves the fluid and wheel at rest (as far as rotation is concerned). The motion of the box, therefore, relative to the wheel, gives the amount of the roll. This indication of the magnitude of the roll would be perfectly accurate, except for friction and viscosity errors, were it not that, in order that the wheel may have a definite zero position, it is necessary to fix a small weight on its circumference, thus converting it into a pendulum.

As before mentioned, a pendulum hung anywhere (except at one particular spot) in a rolling ship will not remain vertical, and it will be of interest to show what the motions of the pendulum are, and how they depend on the length of the pendulum and the period of the roll. When the motion of the pendulum is not resisted by viscosity, a very simple construction gives the required relation.* Let O be the centre about which the ship is rolling, and Q the point of support of the pendulum; let $QR = \lambda$ be the length of the pendulum, and its period $t_2$, also let the period of one complete roll be $t_1$, $\theta$ the maximum angle of roll (off the vertical), and let $OQ = r$. It is required to find the angle $\phi$ which QR makes with the vertical at any instant during the roll. Since $t_1$ is the period of the roll, the pendulum, whose natural period is $t_1$, will have a length $\frac{g t_1^2}{4\pi^2} = l$, say. In the vertical through O take $R^1P = l$. When the ship is inclined, $\theta$, $QQ^1 = r\theta$, and if a pendulum of length $l$, whose point of support is at P, and fixed, be made to swing with an amplitude $\phi$, such that $(l - \lambda)\phi = QQ^1$, the




* In all the subsequent discussions the rolling is supposed to be isochronous, which is equivalent to the assumption that $\theta$ may be used for $\sin\theta$.

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point distant P Q[1] from P on the long pendulum will occupy the same position at every instant of the roll as the point Q supporting the pendulum fixed in the ship.

Hence, since the force on the bob of a pendulum acts in the direction of its point of support, and the point distant P Q from P on the long pendulum remains throughout the roll identical with the point of support of the ship's pendulum, it is evident that that part of the long pendulum Q P which projects below Q behaves exactly as the short pendulum which Q supports. Or, in symbols—

$$\phi = \frac{r \cdot \theta}{l - \lambda}. \qquad (1)$$

There are three cases which it is worth while to examine :—

(1) Suppose, first, that $\lambda$ is very small compared with $l$, then $\phi = \frac{r}{l}\theta$ approximately.

(2) If $l$ and $\lambda$ are comparable in magnitude, $\phi$ becomes large.

(3) If $\lambda$ is large compared with $l$, $\phi = \frac{r}{\lambda}\theta$ approximately, which is a small quantity compared with $\phi$ in case (1).

In case (1) the pendulum will hang in the direction of the force compounded of gravity and the lateral acceleration of its point of support. This latter is partly due to the lateral acceleration given to the ship as a whole by the waves she is rolling in, and partly to the height above or below the centre of gravity of the ship at which the pendulum is hung.

In case (2) the pendulum will acquire large motions of its own from the motion of the ship, and will thus be useless as a Rolling Indicator.

In case (3) the pendulum hangs nearly truly vertical while the ship rolls, and will, therefore, give an accurate measure of the roll.

As an example, suppose the angle of roll to be 20°, the ship to have a rolling period of 10 seconds, and the pendulum to be supported 10 ft. above the centre of gravity. The length of a pendulum whose period is 10 seconds is 81 ft., about, hence—

$$\phi = 20^\circ \frac{10}{81 - \lambda}. \quad \text{If}$$

(1) $\lambda = 1$ ft. $\phi = 2{\cdot}3^\circ$ ;
(2) $\lambda =$ about 80 ft., $\phi$ becomes very large.
(3) $\lambda = 160$ ft. $\phi = -2{\cdot}5^\circ$.
$\lambda = 281$ ft. $\phi = -1^\circ$.

It will be seen that, in the supposed case, however short the ship's pendulum, the error in its indication will never be less than about one-eighth of the roll, while to

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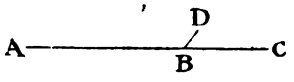
reduce the error to one degree, the length of the pendulum would have to be 280 ft., even on a ship of the comparatively quick period of 10 seconds. If, however, errors of one or two degrees were all that need be feared from comparatively short pendulums, it would be a comparatively unimportant matter, but the result above given is obtained on the assumption that the rolling is regular, and that no free vibrations exist in the pendulum. Neither of these conditions holds at sea. The great difference which the addition of the free vibrations makes in the motion of a system subjected to a harmonic force is illustrated in a paper by R. E. Froude, on "The Non-Uniform Rolling of Ships,"* but a complete discussion of the initial stages of a forced oscillation would occupy too much space to allow of its being given here.†

Without going further into the question of the forced oscillation of ordinary pendulums, I will examine the effect of the forces experienced on shipboard on the Rolling Indicator which forms the subject of this note. If there were no fluid the wheel of the indicator would come under case (1) (that is, it would have a period shorter than that of the ship), and would, therefore, execute forced oscillations about the true vertical when the ship rolled, the period of the forced oscillations being that of the ship, and their phase the same, or opposite to that of the roll. Its indication would thus be wrong by the whole amplitude of the forced oscillation. The presence of the fluid, however, modifies the forced oscillation in three ways. First, by adding very largely to the effective mass of the wheel while slightly diminishing (by flotation) its stability, it greatly lengthens its natural period; secondly, it causes a very rapid extinction of any relative motion of the fluid and wheel, thus reducing to a small quantity the possible amplitude of the forced vibration; and thirdly, which is most important, it alters the phase of the forced oscillation with reference to the phase of the roll.

The following diagram illustrates the effect of this alteration of phase on the indications of the instrument. Let A B be the amplitude of the roll. If A B revolve uniformly round A in the period of the roll, the horizontal component of the motion of B is what the indicator ought to show; but if B C is the amplitude of the forced oscillation, what the indicator will show, in the case of the pendulum without the fluid, will be the horizontal component of the motion of the point C. With the fluid, however, if B D be the amplitude of the forced oscillation (which will, as before stated, be much less than B C), the indicator




* Trans. Nav. Architects, March, 1896.

† If the effects of extinction are neglected, the initial motion of a system whose natural frequency is N, when acted on by a harmonic force of frequency $n$, is $\frac{q \sin\theta - \sin q\theta}{1 - q^2}$ where $q = \frac{n}{N}$ and $\theta = 2\pi N t$.

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will show the horizontal component of the motion of D, which differs very little from that of B.

In symbols, $t_1$ and $t_2$, as before, being the period of the roll and indicator wheel, if a couple $F \frac{\cos 2\pi}{t_1} t$ act on a system whose co-efficient stability (*i.e.*, the restituent couple at unit angular displacement) is $f$ and period $t_2$, subject to a retarding couple of $p$ times its angular velocity, the resulting forced oscillation is represented by

$$\frac{F \cos \beta}{f} \frac{t_1^2}{t_1^2 - t_2^2} \cos \left(\frac{2\pi}{t_1} t + \beta\right) \qquad (2)$$

where—

$$\tan \beta = m \frac{t_1 t_2}{t_1^2 - t_2^2} \qquad (3)$$

and—

$$m = \frac{2\pi p}{f t_2}.$$

Here $m$ is the ratio of the resistance which the system would experience when passing through its zero position with the speed due to unit displacement, stability co-efficient $f$, and period $t_2$ $\left(\text{viz., } p \frac{2\pi}{t_2}\right)$ to the restituent couple at unit displacement ($f$). The natural period ($t_2$) of the wheel of the indicator in the fluid is generally made about 30 or 40 seconds, and the length of a pendulum of this period is from 750 ft. to 1,300 ft. Without the fluid the period of the wheel is about four seconds, and the length of the equivalent pendulum about 13 ft. Hence the added mass of fluid, which by the floats of the wheel is caused to take part in the oscillation, has the effect of increasing the length of the equivalent pendulum more that sixty fold.

If the indicator is placed at a height $r$ above the centre of rotation of the ship the value of F for a roll of $\theta$ is $f \frac{4\pi^2 r \theta}{t_1^2 g}$, hence the amplitude of the forced oscillation is, from (2)—

$$\frac{4\pi^2 r \theta}{t_1^2 g} \cos \beta \frac{t_1^2}{t_1^2 - t_2^2}. \qquad (4)$$

It will be seen at once that this agrees with the expression given in (1) for the amplitude, since that part of the applied force which is required to alter the period of the system from $t_2$ to $t_1$ is F cos $\beta$; hence, since—

$$\frac{4\pi^2}{t_1^2 g} = \frac{1}{l} \text{ and } \frac{t_1^2}{t_1^2 - t_2^2} = \frac{l}{l - \lambda},$$

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(4) becomes—

$$\frac{r\,\theta\cos\beta}{l}\,\frac{l}{l-\lambda} = \frac{r\,\theta\cos\beta}{l-\lambda}.$$

The angle $\beta$ is found by experiments on the rate at which free oscillations of the wheel die away.* Its value is about 30°, so that $\cos\beta = \cdot 85$. Taking, therefore, as before (as an example), $l = 81$ ft., $\lambda = 1{,}000$ ft., and $\theta$ (the angle of roll) $= 20°$, the forced oscillation of the wheel will be $-20° \cdot \frac{r}{1{,}050}$; hence, for the forced oscillation to exceed a degree, the indicator would have to be nearly 60 ft. above the centre of gravity. If the period of the roll, *i.e.*, of the ship, was 20 seconds (which is greater than the period of any existing sea-going ship) $l = 320$ ft. nearly, and a 20° roll would cause a degree of forced oscillation when the indicator was about 26 ft. above the C.G. of the ship.

The above remarks show that in any practicable position on board any actual ship the forced oscillation due to the lateral travel of the instrument in rolling is a small fraction of a degree.

Thus far only the effect of forces due to the extinctive action of the fluid and to causes outside the indicator have been considered.

It remains to examine the magnitude of the oscillation which the viscous drag of the fluid, set in motion by the sides of the indicator box, can maintain in the wheel, and also the effect of the friction of the pivots.

If a mass of viscous fluid, otherwise unbounded, be limited in one direction by a plane, and this plane oscillates parallel to itself, the effect on the fluid is that waves, in which the particle-displacement is parallel to the motion of the boundaries, are propagated from the plane into the fluid in a direction normal to the plane.† The velocity with which these waves travel is proportional to the square root of the viscosity of the fluid, and inversely proportional to the square root of the period. The amplitude of the motion decreases rapidly with the distance from the moving plane falling to $e^{-2\pi}$ or $\frac{1}{535}$ of its original value in one wave length.

In water the wave length is $\cdot 184\sqrt{t_1}$ in. Thus, with a period of 10 seconds, the motion at half an inch within the fluid is less than $\frac{1}{500}$ of the motion of the surface, and with a 20-second period the same reduction takes place in about three-quarters of an inch.

It is plain, therefore, that no rotation of the indicator box due to rolling will

* *See* Maxwell, Elec. and Magnetism, Vol. I., Chap. XVI.

† *See* Stokes' Camb. Trans., 1851, or Lamb, Hydrodynamics, pp. 538, 539.

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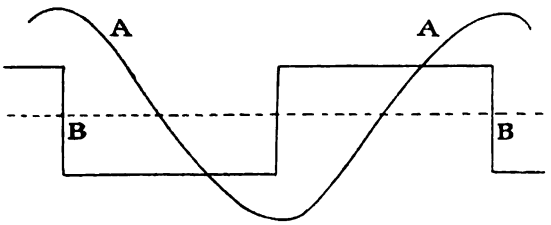
produce by viscous action a rotation of the fluid which is more than a very small fraction of a degree at the distance of an inch from the sides.

Pivot friction is the most important element in causing such small errors as are found to exist in the readings of the indicator. The force between the pivot and its bearing is proportional to the load on the pivot, but independent of the angular velocity. The friction always tends to turn the pivots in the direction of motion of the box. Hence, while the ship rolls from port to starboard there is a constant couple tending to turn the wheel in the same direction, and *vice versâ*.




If the curve A A represent the rolling, the frictional couple acting to turn the wheel will be represented by the curve B B. If the rolling A A is represented by $\theta \cos \frac{2\pi}{t_1} t$ and B B is expressed (by Fourier's theorem) in a trigonometrical series, the function which represents the force indicated by B B is—

$$\frac{4}{\pi}\left(\sin \frac{2\pi}{t_1} t + \frac{1}{3} \sin \frac{6\pi}{t_1} t + \frac{1}{5} \sin \frac{10\pi}{t_1} t + \&c.\right),$$

where $c$ is the couple on the wheel due to the friction of the pivots.

Each term in this series will set up a forced oscillation of the wheel whose amplitude can be determined by (1) or (2). Thus the amplitude of the oscillation occasioned by the first term is, in circular measure—

$$\frac{4c \cos \beta}{\pi f}\left(\frac{t_1^2}{t_1^2 - t_2^2}\right).$$

Of the second—

$$\frac{4c \cos \beta}{3\pi f} \frac{t_1^2}{t_1^2 - 9t_2^2},$$

and so on, the amplitudes due to the successive terms rapidly diminishing.

It will be sufficient for the present purpose to consider the effect of the first two terms.

The quantity $c$ is found by experiment to be about $\cdot 01 f$. The values of $\beta$ will be different for each term, $\frac{1}{3} t_1$, $\frac{1}{5} t_1$, &c., being substituted for $t_1$ in (3).

The influence of the second term is never felt, unless (as in some of the tests applied to the indicators) the applied couple has nearly three times the natural period of the indicator wheel. In this case the second term causes measurable oscillations, but it has no practical importance, waves of a hundred seconds period not being met with at sea,



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When rolling first begins, or when it is irregular, free vibrations are set up as well as the forced vibrations above discussed, and this is true as regards the wave force as well as pivot friction. In the initial stages of the change, the sum of the natural and forced vibrations may be as much as a degree.

It will be observed that the forced oscillations, due to the wave forces, differ in phase from the roll by the angle $\beta$, while the forced oscillations, caused by the pivot friction (which would be 90° different in phase from the roll if the fluid in the box had no extinctive effect), differs by $90 - \beta$ from the phase of the roll. Thus, since the rolling observations are made when the inclination of the ship is a maximum, *i.e.*, when the phase of the roll is $o$ or $\pi$, the magnitude of the error introduced by the presence of a forced oscillation, $\alpha$, due to wave forces will be $\alpha \sin \beta$, while a forced oscillation, $\gamma$, due to pivot friction will, at the same instant, be represented by $\gamma \cos \beta$. Hence, though the alteration of phase has not much effect in reducing the forced oscillation due to lateral acceleration, it, in a large measure, eliminates the more considerable error due to pivot friction. The sum of these terms will rarely equal a degree in any ship with less than a 20 second period, and, in general, it will be less. Since the forced oscillation, caused by pivot friction, is independent of the angle of rolling, the percentage accuracy of the observations increases with the magnitude of the roll; but, for this reason also, the greatest care has to be taken to make the bearings as frictionless as possible.

In certain cases pitching may tend to increase the frictional forced vibration. The effect of pitching on the fluid in the indicator box is to cause a flow of the fluid to take place in the sense opposite to that in which the box turns about the axis of pitching. When the pitching exceeds a certain amount, it intensifies the pressure between the wheel pivots and their bearings; but experiments with the indicators show that this effect is not sensible unless the pitching is abnormally great.

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## DISCUSSION.

The Secretary read the following remarks contributed in writing by Mr. R. E. Froude :—

Mr. R. E. Froude, F.R.S. (Associate Member of Council): I regret my inability to be present at the reading of Mr. Mallock's paper. The instrument he describes stands alone in supplying something greatly and long needed. Setting aside the bulky, elaborate, and very costly instruments which have occasionally been used for special rolling experiments, Mr. Mallock's Indicator is the only existing appliance by which, when batten observations are not practicable, the rolling of a ship at sea can be observed with even colourable correctness. And, even when the

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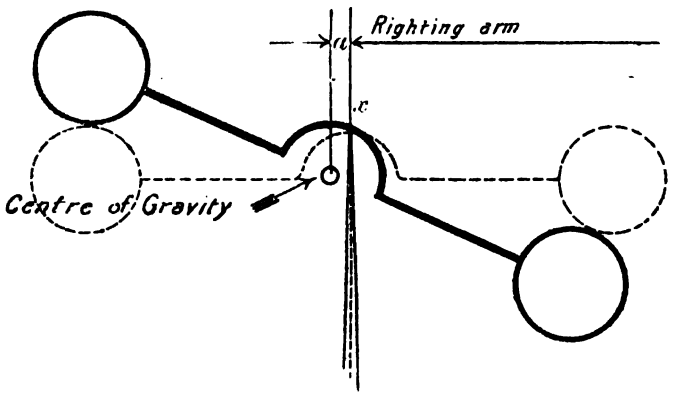
conditions appear to admit of batten observations, their use is not only less convenient than that of Mr. Mallock's Indicator, but their indications are not, in my experience, always quite trustworthy. For rolling angles of any magnitude, separate observations on the two sides of the ship are requisite, and then any defect in apparent horizon becomes a direct element of error in the measurement of the out-to-out roll. I may add that, preparatory to the use of one of Mr. Mallock's Indicators, in some rolling experiments which I am about to conduct for the Admiralty, I have carefully studied Mr. Mallock's theoretical investigation of the limits of error of the instrument, and have also subjected the instrument itself to a series of experimental tests. I have found the investigations to be not only perfectly correct, but to be also verified in an interesting way by the results of the tests.

Mr. C. H. Wingfield (Member): The author suggests a long-period pendulum several hundred feet long. In a paper written for this Institution many years ago by Mr. Thornycroft he described an instrument, not for ascertaining what the rolling angle of a ship was, but to stop it rolling at all. He had a very interesting automatic machine for this, in which the two essential parts were a long-period pendulum and a short-period pendulum, and the long-period pendulum was made thus: a long lever with a weight at each end, so that it had a large moment of inertia. This was supported at $x$, which was very slightly above the centre of gravity, so that the righting arm $a$ was short. Consequently a pendulum with as long a period as you like can be in this way made to occupy a very small space. I only mention this as a matter of interest. An experiment due to Mr. Hopkins, an American exponent of popular science, bears out what the author demonstrates mathematically about fluid not having much motion communicated to it by the containing vessel. He says you can show the rotation of the earth very well (I have not tried the experiment, but I believe it is quite correct) by taking a basin of water, sprinkling lycopodium on the surface, and then with some black dust drawing a diametrical line across the lycopodium, and marking its then position on the basin by laying a walking stick parallel to it. In six hours or so you will find the water has stood still while the basin rotates with the earth, as shown by the walking stick lying at 90° to the black line, with which it was at first parallel.



Long-Period Pendulum.

Professor G. H. Bryan, M.A., F.R.S. (Visitor): This paper has interested me considerably. The writer has pointed out that in observing the oscillations of a ship, pendulums are of very little use. The reason may be summed up in a few words, by saying that when a pendulum is attached to a ship it does not remain vertical, but oscillates; and it is very difficult to find out exactly what the motion of the ship is from observing the oscillations. The last speaker has pointed out that you can give a pendulum as long a period as you like by making the moment of inertia large, and making the distance of the centre of gravity below the axis of support small. Such an arrangement, instead of always oscillating, is very liable to be set continuously rotating. There are several ways in which

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the rolling of a ship could theoretically be observed. Mr. Mallock has referred to the use of battens. Then, again, we might observe a star at night which would give an absolutely fixed direction, and, if one could have an instrument for focussing the image of the star on a moving photographic plate, a perfect arrangement for observing the rolling of ships would be secured. A further method would be by the use of the gyroscope, an instrument which has been used, moreover, for the purpose of demonstrating the rotation of the earth. I must say that I am a little bit sceptical about the arrangement of water and lycopodium used for proving the earth's rotation, because the water that you are observing necessarily acquires in time the same angular velocity as the earth in consequence of its viscosity. Now, let us consider the difference between Mr. Mallock's arrangement with fluid inside, where the inertia of the fluid is added to the inertia of the paddles, and a pendulum which is made very long by increasing its moment of inertia. In the case of the pendulum, the principal cause tending to retard the oscillation is the friction of the pivots plus a small amount of air resistance. In this arrangement with fluid viscosity, the fluid tends to retard free oscillations, and very soon to extinguish them. Now there is one great difference between the viscosity of the fluid and the resistance due to friction of pivots. As commonly assumed, the resistance due to friction of solids is independent of the velocity, and merely changes from positive to negative as the direction of the motion is changed. If the friction of the pivots were considerable, and the motions of the ship slow, the paddle would simply turn with the ship, and the instrument would give no readings. On the other hand, the resistance due to viscosity is proportional to the relative velocity, and its action tends to extinguish the free oscillations of the paddle, as well as to prevent the latter from being set in continuous rotation, never to completely stop the relative motion of the paddle and prevent the instrument from giving readings. Probably this apparatus practically serves its purpose as well as any simple contrivance that could be designed. The friction of the pivots seems the chief drawback, but it would appear that the effects of this cause can be made small, even when account is taken of the fact that movements of the ship, which tend to cause the paddle to press on the pivots, may increase the friction, as pointed out in the paper. I should like to inquire whether experiments have been made on the use of a gyroscope for observing the rolling of ships. At the same time, it does seem to me that this instrument fulfils all the requirements. It would be interesting to know whether Mr. Mallock proposes to have the instrument fitted with an automatic recording apparatus. There would be no difficulty about this. It could be done, for example, by having a mirror arrangement projecting a photographic image on a moving plate.

Mr. F. J. Trewent (Member): My Lord and Gentlemen, I have listened with very great interest to Mr. Mallock's paper. This is a very ingenious instrument. The author has told us that the natural period of the wheel pendulum in the air is 4 seconds, when it is immersed in water it is about 30 to 40 seconds. This is an enormous alteration in period, and I think it requires a little examination. Mr. Froude in his wheel pendulum also obtained a period of about 34 seconds. Of course, this was a very heavy wheel pendulum, about 200 lbs., and I think, if I remember rightly, his centre of gravity was about $\frac{6}{1000}$ below the point of suspension. I understand Mr. Mallock's pendulum, instead of this great weight, is only a few ounces. Now, by what subtle means does he obtain this long period? I think it is simply obtained by appropriating the inertia of the water throughout the diameter, and the width of his wheel. It is practically the same as if it were a cylinder wholly immersed in water of the same weight, with the centre of gravity slightly below the axle. Such an apparatus as this was experimented on by Mr. John A. Rowe many years ago.

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Mr. Rowe, it will be remembered, succeeded Mr. Macfarlane Gray as Chief Examiner of Engineers to the Board of Trade in 1897. He experimented with instruments almost identical with Mr. Mallock's, and completed his experiments, in 1893, by adopting a heavy cylindrical float of exactly the weight of water displaced, and having the centre of gravity an exceedingly small distance below its centre of buoyancy. He used it to solve the very difficult problem of firing guns at sea. Those gentlemen who have had connection with the North-East Coast Institution will have heard him several times refer to that in years gone by. In his case the cylinder weighed several hundred pounds, and was not a light apparatus of this kind. He also obtained a period of from 30 to 40 seconds. Mr. Rowe has, I believe, had this business in hand for the last twenty years. I quite agree with Mr. Mallock that when there is reasonable water space between the cylinder and the outer case, no motion at all will be conveyed from the outer moving case to the wheel pendulum in any way. I do not quite agree with Mr. Mallock as to the point of friction. It seems to me that he has made far too much of this. If the body is immersed in water, and is made exactly the same weight as the water, there is no friction at all, because the body is actually supported, and there should be no friction. The body then practically possesses no weight, it is the same weight as the water which it displaces. Mr. Mallock, at the end of his paper, says:—"In certain cases pitching may tend to increase the fractional forced vibration." I scarcely see why this should be so, except in vessels of very short pitching periods. In battle-ships and large cruisers, the anticipated tremble may be ignored. To violence of rise and fall, to sudden reversals of speed ahead and astern, an instrument such as Mr. Mallock's or Mr. Rowe's is, so far as friction is concerned, utterly indifferent, and it is certain that no pressure is exerted on the pivot bearings through the violence referred to. Some time ago I obtained a clear bottle and filled it with fresh water. In this I placed a cylindrical-shaped piece of wood exactly the weight of water displaced. With a little manipulation it was placed in the centre of the bottle, where its slightest motion was visible. I then shook the bottle as rapidly and violently as possible, but the immersed wood cylinder was absolutely motionless. If the weight were increased beyond that of the water displaced, or decreased below that of the water displaced, motion would instantly occur. The secret which makes this invention so admirable was discovered by Mr. Rowe in the above-named year, and consisted of making the immersed object, cylinder or otherwise, of precisely the same weight as the water displaced. The reason why the friction observed by Mr. Mallock in his instrument occurs in pitching motions is because the indicator is not free to turn about its centre of buoyancy, to bring its centre of gravity vertically under it. The pivot bearings prevent this, but the force on the bearing, necessary to make the immersed object partake of the pitching motion of the vessel, will, in the worst of cases, be only a few ounces, even when the immersed body weighs several hundred pounds, and even this friction may be obviated. I do not think I have anything further to add, except to thank Mr. Mallock for his very interesting paper.

Mr. Philip Watts, F.R.S. (Vice-President): My Lord, I desire to say that my firm* has supplied these instruments to the ships which it has turned out for many years past, with very satisfactory results. They have continued in good condition, and have given the roll of the ship accurately. I believe it is the only instrument in the market which does this. The roll has been measured with the gyroscope, as the last speaker but one suggested, and I have no doubt it could be measured by means of a star or any other distant object. This is practically the method referred to

* Sir W. G. Armstrong, Whitworth & Co., Ltd

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by Mr. Mallock as the method by "batten observations," where the horizon is the distant object. But the horizon cannot always be seen, nor can a star or other distant object always be seen; besides which the arrangements and mechanism for observing by means of a distant object, as well as by a gyroscope, are cumbersome or complicated, or both; whereas Mr. Mallock's instrument is exceedingly simple. It is always in place, it may be put anywhere; it is always measuring the roll, and is ready to be observed. I have invariably heard it spoken of very highly by those who have used it.

Professor J. H. Biles (Member of Council): My Lord, I think that the question of observations of rolling and the inclination of a ship at sea has not received quite as much attention as it might have. It is probably quite within the experience of most here to have results of observations stated which are obviously absurd. These results are taken from pendulums which are hung up in all kinds of places in a ship, and the records which are given by these pendulums are anything but the real facts of the case. I have in Glasgow an apparatus which represents a rolling ship, and a pendulum is placed on that apparatus at various heights. It is quite easy to adjust the periods of oscillation, the periods which represent the rolling of the ship, and the period of the pendulum, so that, when the ship is set rolling, the pendulum makes a series of complete revolutions and continues to revolve as long as the main body oscillates, showing that the pendulum, if put in an improper place, may have really no rotation whatever to the real angle of inclination of the ship. I think the fact that this is not known widely by shipowners, ship captains, and others who go to sea in ships is the reason that we have not had a more urgent demand for an apparatus such as Mr. Mallock proposes. I am sure this apparatus will be of very great interest, especially if it is modified—supposing it has not already been—in the way Professor Bryan mentioned, by having a recording apparatus attached. This would be most interesting, as the actual rolling of the ship at sea may be automatically recorded, and brought home to those who do not go to sea frequently. They might then have some reliable data for studying the actual rolling of ships at sea. This paper of Mr. Mallock's, if I may be allowed to say so, is a most interesting and ingenious one, and shows, I think, clearly that this apparatus ought to do its work well. We have Mr. Froude's dictum that, from his observations, it does its work well, and it is one of those curious instances where, unfortunately, practical experience is not the ultimate test, because we have no real datum line upon which to measure the rolling of ships. Unless this instrument is accurate in its records, we have no means of saying that those records are correct. We may, perhaps, take casual observations by a batten instrument; but it is an instrument on the accuracy of which we have to rely, and we cannot very well check it by any easy means. It has, therefore, been necessary for Mr. Mallock to write this paper in fuller detail; not, as he put it, giving us a very dull paper. I think it a most interesting paper, and it is interesting because he had to justify the accuracy of this instrument, for the reason it could not justify itself in any other way than by his investigation.

Mr. A. Mallock (Visitor): I do not see there is very much to reply to in what has been said. As regards the long pendulums mentioned in the paper, the lengths referred to are the lengths of the simple equivalent pendulums having the given periods. The same periods can be got by giving a short pendulum a large moment of inertia. This wheel is an example. In air it would be a pendulum of short period, but it is loaded with the water, so that its effective mass is very large, although the weight on the pivots is not increased, in fact, is diminished by the water. There the

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restituent force is really the eccentric weight. Here you have an example of a pendulum where the solid mass is small, but the mass which is compelled to move with the pendulum is large and the period very long, because the restituent force is very small. Somebody referred to Mr. Froude's pendulum. Mr. Froude has a wheel for measuring rolling. That is the sort of thing you would go to for really accurate measures of rolling. It is a beautiful example of refined workmanship and design, but is entirely unfit to be trusted out of the hands of skilled people. When you want real accuracy, an instrument of that sort should be employed, and its indications treated by proper analysis afterwards. There would be no difficulty in adding a photographic recording apparatus to such an indicator as I have here; and I dare say for sea-going purposes it would be worth while to do so. The friction of the pivots was referred to. I have gone into that matter very carefully, and its effects are known to me by elaborate experiments. I find that pivot friction is the most important factor in setting up oscillations of the wheel. The fact that the flotation takes nearly the whole weight off the pivot does reduce the friction enormously, but it does not altogether eliminate it.

Captain H. J. May, R.N., C.B. (Visitor): I suppose, in the small instrument, your weight being so light, your friction is a very large proportion of the whole thing?

Mr. Mallock: Yes, the friction is exceedingly small, but the forces required to set up these small oscillations are also so small that one has to look on all friction with a most watchful eye. I have nothing more to say, except to thank you for your very kind reception of my paper.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): I am sure you will all join with me in a very hearty vote of thanks to Mr. Mallock for his very interesting paper.

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# ON SOME EXPERIMENTS MADE ON BOARD THE ATLANTIC LINER "DEUTSCHLAND," DURING HER TRIAL TRIP IN JUNE, 1900.

By Herr Otto Schlick, Member.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 28, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

The principal cause of vibrations in steamers, so far as these consist in the bending of the ship's axis in a vertical plane, is well known to lie in the up-and-down action of the moving parts of the engine. To prevent these vibrations it is absolutely necessary to use an engine with balanced forces, or, as it is usually termed, a balanced engine. It has been found that this precaution is not only necessary for men-of-war, and passenger ships of proportionally high engine-power and speed, but even in cases of large cargo steamers of medium speed. On account of the varying displacement of this latter type of vessel, the so-called "critical number of revolutions" varies to such an extent that it is not always possible to arrange the number of revolutions of the engine in such a way as to prevent vibrations, which under circumstances have proved to be a great drawback to the rigidity of this class of ship.

It has repeatedly been found that even a vessel provided with a carefully balanced engine has still shown slight vibrations at the critical number of revolutions of the engine, although in a considerably less degree than in vessels with unbalanced engines. I particularly emphasise, that only vibrations are dealt with here, the period of which coincides with that for one revolution of the engine.

Most of the vessels supplied with my system of balanced engines have been run for trial with the propellers uncoupled to examine the effect of the balancing. On all these trials vibrations have never been noticed, even when the engine reached the critical number of revolutions; whereas engines not balanced always show excessive vibrations when running with disconnected propellers, as soon as the critical number of revolutions is reached.

The results mentioned here, in short, prove the efficacy and importance of the balancing of forces, but they also prove that the cause of vibrations in vessels with balanced engines does not lie in the engines themselves, and must be sought elsewhere.

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It has been tried, especially in Germany, to attribute the cause of the vibrations in steamers with balanced engines to the uneven rotation of the engines, and, in consequence, of the propellers also. It was argued that the variations existing in the speed of rotation of every engine cause a jerking effect in the pressure given by the propeller, and that this pressure acts like the power exerted by a rower in a boat. As the direction of the pressure produced by the propeller does not go through the centre of gravity of the ship—according to the views of the defenders of this theory—a bending of the ship's hull in its longitudinal axis must take place, which produces vibrations when regularly repeated.

There are a great number of arguments which prove the incorrectness of these statements, but it would be going too far to enter into all of them here. I will, therefore, only state, as an easily understood proof to the contrary, the fact that vessels with engines of great variation in their speed of revolution have repeatedly shown no measurable vibrations; while, on the other hand, vessels with an exemplary even-turning speed showed considerable vibration.

To investigate the real cause of the vibration in vessels with balanced engines, I considered it necessary to make accurate experiments as to the position in which the cranks of the two engines are to be found when the vibratory movements are in a certain point of the swinging track, *e.g.*, when the axis of the hull of the ship is in its mean position. The results of these experiments promised far-reaching disclosures as to the circumstances accompanying these vibrations, which up to now have quite escaped our notice.

Through the kindness of the Stettiner Maschinenbau-Actien-Gesellschaft "Vulcan" I was enabled to make the before-mentioned experiments on the trial trip of the steamer *Deutschland*, and the results of these experiments are shown in what follows.

After thoroughly examining the methods to be taken into consideration for gaining the proposed end, and after similar trials had been made on another vessel, it was agreed to proceed in the following manner.

An electric contact was fixed on the shaft of each engine in such a way that the current of six strong cells closed when the forward crank of the engine was still 30 degrees from the vertical position, shown later in the drawings as crank I, and that it opened when the crank was exactly in its top position. The contact therefore took place during the time the crank required to move through an angle of 30 degrees. The current was conducted through the primary reel of a powerful induction coil, and gave forth on opening and closing a strong spark, which proceeded from a pointed metal pin just above the paper band on to the metal roller underneath, thus burning a small hole



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in the paper. The marking on the paper therefore took place instantaneously. Such an arrangement was made for each engine, so that the revolutions of each engine were marked on the paper band. The marks of the starboard engine are on one side, and those of the port engine on the other side, of the paper band.

The idea of producing the marks on the paper by means of the secondary current is that of Herr Dr. Bauer, an engineer at the "Vulcan" Works, and I am also greatly indebted to him for the energetic assistance he rendered me in carrying out these experiments.

The diagram of the Pallograph showed a picture as in Fig. 1, Plate XXXV., which represents its natural size.

It must be expressly mentioned here, in order to understand what follows better, that the Pallograph was placed at the extreme after-end of the steamer, where the strongest vibrations are always felt.

In each of the diagrams shown here, the top parts of the wave-line of the vertical vibrating curves always correspond to the movements when the swinging of the after-part of the ship is above her mean position, and the lower parts of the wave-line correspond to the movements when the after-part of the ship is in her lower swinging position.

It will be noticed that there are always two marks made by the electric current, and which are close together. The one to the left is always that made first, and represents the spark caused by the closing of the current. The second, which is always somewhat more marked, is caused by the opening of the current; and, as the opening always takes place suddenly, on account of the peculiar contact arrangement, and acts at a fixed point of the periphery, the mark made by the opening spark is only taken into consideration in the following experiments.

In order to understand the diagram drawn by the Pallograph correctly, it will be as well to assume next that the paper band, which usually unwinds at the rate of 1 in. per second, is at rest. The marks made by the two pens which indicate the vertical and horizontal vibrations, the mark of the pen which shows the seconds, and the marks made by the electric current will then be found to be exactly in a straight line at right angles to the edge of the paper band.

In order to ascertain in what position of the up and down movement of the ship, crank I (that is, the forward crank) stands vertically upwards, it is necessary to draw straight lines, at right angles to the edge of the paper, through the marks made by the electric sparks on the diagram. These lines always represent the moments when the respective crank is standing vertically upright.

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Those representing the marks of the starboard engine are drawn with green ink in the original diagram, and those of the port engine are in red ink.

The diagrams then assume the appearance as seen in Fig. 2 of Plate XXXV. To simplify matters, the figure has been printed in black only, so that the cross-lines which represent the marks of the port engine, and which are in red in the original, are here drawn out in full, and those which refer to the starboard engine, green in the original, are represented by dotted lines.

We can now, for instance, conclude from the diagram of Fig. 2, Plate XXXV., that the starboard engine lags behind the port engine, or, in other words, is revolving slower. At point *b* both cross lines fell almost exactly together, *i.e.*, the cranks of both engines were in symmetrical positions, both cranks I were going through the upper dead point at the same moment.

After 7 revolutions the starboard engine is already a distance of $h''$ $h'$ behind, and it is now important to know how much this difference is expressed in degrees. As the paper band cannot be made to move absolutely uniformly, a proportional scale is used which is divided into 360 parts. We take the distance $h'$ $i'$, equal to 360°, and we can then read the angle made by the distance $h''$ $h'$ direct off the scale; by this means we can see, at any part of the diagram, how much the crank I of the one engine is ahead of the other.

In a similar way, by means of the time marks, the exact number of revolutions can be ascertained which each engine makes within a certain time, for instance, in one minute; and 1/100 part of a revolution per minute can be read with approximate accuracy, if the number of revolutions does not much exceed 80 to 100.

It may be mentioned here that the results of these experiments have shown that there is a continual slight variation in the speed of the revolutions in each of the two engines of a steamer. Probably these are caused by the rudder being slightly shifted, which is unavoidable even when steering a straight course.

From what has been said, it will be seen that, with the assistance of the arrangements described, we are in a position to determine the following particulars from the diagram at any moment of observation, viz.:—

The turning speed, or number of revolutions, of each engine;

The angle at which the crank I of the one engine is ahead of the other;

The position of the two cranks I during every phase of the vibratory movement of the hull of the ship; and, besides this, the Pallograph gives the number, form, and the extent of the vibrations.

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Of the diagrams taken during the trial trip of the *Deutschland*, on June 26 and 27, 1900, nine of them proved to be useful for analysing; namely, six diagrams (No. I. to No. VI.) which had the marks made by the induction current, and three diagrams (Nos. VII. to IX.) without these marks.

It must be remembered, on examining these diagrams, that the vertical vibrations, by making use of the lever on the Pallograph, appear twice the size they are in reality, in order to gain a clearer picture. The horizontal vibrations on the Pallograph are given in their natural size.

The results worked out from these diagrams are arranged in the Table given at the end of this paper.

The maximum amplitude of vertical vibrations at the extreme after end of the ship was about three-sixteenths of an inch. One must acknowledge that this is an exceedingly small amount for a ship of 662 ft. in length and 37,000 horse-power.

## VERTICAL VIBRATIONS.

We will now proceed to examine the vertical vibrations, *i.e.*, those vibrations which consist in the bending of the ship's longitudinal axis in a vertical plane.

It will first be of interest to determine the critical number of revolutions for the vertical vibrations.

Diagrams I., II., and VII. were used to determine the critical number of revolutions. On the whole the vertical vibrations show little regularity, so that it was very difficult to determine the maximum amplitude.

The lowest number of revolutions noticed will be found on Diagram I., Positions I. and II., with port engine 62·83 and starboard engine 62·81 revolutions (see Fig. 3, Plate XXXV.). The vertical vibrations in this case are about one-sixteenth of an inch.

On Diagram II., the greatest amplitude, viz., three-sixteenths of an inch, will be found at position VIII., as represented in Fig. 4, Plate XXXV. The revolutions are for port engine 66·9, and starboard engine 67·2, on an average 67 revolutions.

There are, indeed, in some parts vibrations of greater amplitude, but they cannot be taken into account here, as they are apparently caused by the oscillations of the weight of the Pallograph itself. Only those vibrations which appear regularly are taken into consideration.

Diagram VII., Plate XXXVII., which has no electric marks, shows 22 vibrations in 20 seconds exactly at Position I., therefore, 66 vibrations of $\frac{6}{32}$ in. amplitude per

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minute. There are, however, other circumstances to be considered, to which we will refer later, from which one may conclude that the critical number of revolutions lies at about 67.

The critical number of revolutions estimated when designing the ship was N = 68·7, according to my formula—

$$N = \phi \sqrt{\frac{T}{D L^3}};$$

it, therefore, deviates perceptibly from the number realised by these experiments, which may be accounted for by the deck erections added afterwards, whereby the distribution of the weight became different from that which had been estimated when adopting the co-efficient $\phi$.

## Maxima and Minima.

On looking through the diagrams carefully, and comparing the Table given at the end of this paper, it will be seen that a maximum of vertical vibrations always occurs when the port engine is about 90° to 100° ahead of the starboard engine; and a minimum, when the difference in the position of the cranks is a further 180°, or when the port engine leads by 270° to 280°, which is the same as if the port engine is 80° to 90° behind the starboard engine.

In some places the angle of advance differs essentially from 100°, as, for instance, in Position I. on Diagram I. (see Table, p. 62), where the lead is only 55°, and in Position III. on Diagram III., where the lead amounts to 150°. It must, however, here be taken into consideration, for instance, at Position I., Diagram I., that the advance never exceeded 55°, but that the port engine ran slower again. Had the advance reached 90° to 100°, the vibrations would probably have been still more violent. It is somewhat similar at Position III. of Diagram III. Here the advance was at first greater, whereby no vibrations occurred; but when the advance reached 150°, vibrations were already perceptible, which became more violent as the advance decreased. This could not be followed up, because the paper band was used up at this point, and therefore, strictly taken, one cannot here speak of a maximum.

The circumstance that the maximum of vertical vibrations of the first order only occurs when the port engine is 90° to 100° ahead of the starboard engine already enables one to draw some important conclusions.

It is to be seen from this that the vibrations cannot be caused by deficient balancing, nor by the want of uniformity in the turning moment, as has often been supposed. The cause cannot lie in the engines at all; for, if that were the

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case, this cause would always occur in the one engine exactly when the other was in its symmetrical crank position. Therefore, when the maximum is attained, both the cranks I would have to pass the upper dead centre simultaneously; on the diagram the marks of the starboard and port engines would have to correspond to an angle of 0°, and would have to be in one perpendicular line at right angles to the edge of the paper band, or, in other words, the red and green lines would have to fall together. The theory of the influence of the turning moment is, therefore, quite out of question.

The object of further investigations was to find the relationship between the vibratory movements (or the curves drawn by the Pallograph), and the impulse causing the vibrations; or, in other words, to ascertain in which phase of the vibratory movement the ship's hull is to be found when the vibratory impulse has reached its maximum.

In order to do this, we must call into requisition some laws of the theory of vibrations of prismatic rods, which I presume are well known.

When a periodically acting impulse, which, for instance, may vary according to Cosine laws, acts on an imperfectly elastic rod, vibration takes place, the amplitude of which attains its maximum when the period of the impulse corresponds exactly with the period of the rod. The moving direction of those parts of the rod that are under the influence of the impulse does not, however, in this case correspond with the direction of force of the impulse in all points; the phase of the periodically acting impulse runs rather 90° ahead of the phase of the vibratory movement, or, as it is generally termed, a phase-difference of 90° takes place.

This, however, as already stated, answers only in the case of exact synchronism of the periods, *i.e.*, for the maximum amplitude of the vibration.

If the period of the impulse differs from that of the vibrations of the rod itself, another phase-difference takes place, which will be very small with a noticeable difference in the periods, and may usually be disregarded.

The above will already enable us to draw the necessary conclusions from the diagrams.

We will next examine Position III. of Diagram II., which shows a maximum vibration, and is represented in Fig. 5, Plate XXXVI. For this purpose we must determine, by visual observation, the two points *a* and *b*, which correspond with the maximum amplitude, and we can then find point *c* by dividing the horizontal distance of the points *a* and *b*, as well as by dividing the vertical distance of these points.

Point *c* then corresponds with the mean position of the hull of the ship, through which the afterpart of the ship is then moving in an *upward* direction.

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We see by the Diagram that the port engine is 100° ahead of the starboard engine (see Table).

It further shows that, at the moment which corresponds with point *c*, crank I of the port engine is 90° past the top vertical position, and the corresponding crank of the starboard engine, 10° behind the vertical position.

The number of revolutions of both engines at this part of the Diagram amounts to 63·8 for the port engine, and 66·2 for the starboard one; the average, therefore, still differs considerably from the critical number of revolutions. Accordingly the phase difference can only be very slight at this moment, and consequently point *c* will correspond with the mean position of the period of the impulse, or, in other words, the impulse will attain its maximum at the moment which corresponds with point *c*.

We find quite similar conditions in Positions I., II., IV., and V. of Diagram II., which are given in the Table.

If one draws the crank positions corresponding to these particulars, and the relative position of the propeller blades, we get the Figs. $6^a$ to $6^e$ for the different positions of Diagram II.

One must now bear in mind that the positions of the propeller blades which are drawn here, always represent that moment when the maximum of the vibratory producing power is attained, and when the afterpart of the ship is in its mean position and moving in an upward direction.

It will be found that both the blades marked A A are nearly in a horizontal position, and moving in a downward direction. If one assumes, therefore, that both these blades meet with a greater resistance, due to the fact that they have only a little larger pitch or a larger surface than the others, or that the edge resistance is somewhat larger, the noticeable vibrations are at once explained.

It will be noticed that the blades A A do not lie exactly horizontally outwards, but are inclined about 30 degrees, proceeding from the centre outwards. One may, herefore, assume that the blade finds the greatest resistance in the water in this position, and this is quite evident.

In order to understand this, we must next imagine that the water is following the ship with tolerable speed, which is considerably greater at the surface than at a certain depth. When looked at from this point of view, the blade will meet with the greatest resistance in the top position. It must, however, be taken into consideration that the flow of the water to the propeller does not move parallel to the axis of the propeller, but that the water rises at the afterpart of the ship in an upward direction and strikes the propeller at an angle, as shown in Fig. 8, Plate XXXVI. This latter circumstance

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would mean that each blade meets with the greatest resistance when pointing outwards exactly horizontally, and its least resistance when lying horizontally inwards.

Taking into consideration the action of the water following, the propeller blade would meet with the greatest resistance at the top, and considering also the rising flow of the water to the propeller, the blade would meet the greatest resistance in an outward horizontal position. As both these influences act simultaneously, the conclusions drawn from our experiments are correct, according to which the blade finds its greatest resistance, and produces its greatest reaction upwards, when rising about 30° outwards, as shown in Positions A and $A_1$, Fig. 9, Plate XXXVI.

The position of the blade when turned 180°, falling inwards 30° from the centre, is that in which the blade produces its greatest reaction downwards, as shown in Positions B and $B_1$, Fig. 9, Plate XXXVI.

The correctness of the conclusions drawn from the results of the experiments is also confirmed by the appearance of the vibrating minima. We have seen by the diagrams that a minimum takes place when the one engine is 270° to 280° ahead of the other, that is, 180° more than at a maximum.

If one assumes that, in Fig. 9, Plate XXXVI., the blades A and $A_1$ have a somewhat larger pitch, and that the port engine is running ahead, the position of the blades drawn in Fig. 10 is that at which the minimum of the vibrations takes place; for, the blade $A_1$ produces just at this moment its greatest reaction upwards, and the blade A its greatest reaction downwards. Both forces nearly neutralise each other, and, as seen from the Diagram, the vibrations completely disappear: this is also a proof that there are no other forces which produce vibrations of the first order.

The position of the propellers at the minimum of the vertical vibrations is as shown in Fig. 10, or, if the two propellers have revolved 180°, as shown in Fig. 11, Plate XXXVI.

In the Positions VI., VII., and VIII. of Diagram II., which all represent the maximum vibration, other relations appear, differing considerably from those we have so far examined. A glance at the Table shows that at Positions VI., VII., and VIII., the cranks I have moved considerably farther from the vertical position, when the after part of the ship is moving upwards through the mean vibratory position.

It is most remarkable that, at Position VIII. of Diagram II., crank I of the starboard engine is here 80° beyond the vertical position, and crank I of the port engine 170°. Both cranks with their propeller blades A and $A_1$ have revolved about 90° farther than they were when the maximum vibration was examined before. The explanations for these appearances are easily understood, if one takes

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into consideration that, in the case of Position VIII., the revolutions of the port engine are 66·9, and the starboard engine 67·2 per minute. The average, therefore, corresponds fairly accurately with the critical number of revolutions. According to the laws of vibrations mentioned before, a phase-difference of 90° must, in this case, take place. The phase of the impulse is 90° ahead of that of the vibration.

From these investigations, therefore, we come to the conclusion that the blades marked A and $A_1$ find a somewhat greater resistance when turning in the water, and that they produce the greatest reaction in an upward direction when rising at about an angle of 30° from the centre outwards. This difference in the resistance of the blades is the only cause which produces vibrations in a vertical plane.

When the minimum of vertical vibrations takes place, some small vibrations are still noticeable, which number 12 per revolution, and are therefore of a very high order, probably 5th or 6th order. The causes of these vibrations cannot be ascertained with any certainty, but probably the impulse of these vibrations is given by the propeller blades, because the number of vibrations during one revolution can be divided by four.

## HORIZONTAL VIBRATIONS.

The horizontal vibrations are attributed to torsional vibrations, as I have already repeatedly explained in other places.

All diagrams show that, during one revolution, four complete torsional vibrations occur. They show the same number of vibrations during one revolution as the propeller has blades.

This is an occurrence which is to be found in all twin-screw steamers, and which must, without doubt, be attributed to the action of each propeller blade. It has always been observed that, when another propeller with a different number of blades was put on, the number of the horizontal vibrations varied accordingly.

According to the Diagram, and the observations taken, the maximum of torsional or horizontal vibrations is attained when the engine is turning at the rate of about 75 revolutions per minute.

It is to be seen from Diagrams II., V., VI. that the maxima and minima of horizontal vibrations also take place regularly. During the period in which one engine leads by one revolution, 4 maxima and 4 minima take place. This occurrence, as will be seen directly, is also to be attributed to the action of the propeller blades. This fact has always been confirmed by the variation which takes place in the number of the maxima and minima when the propellers are replaced by others with a different number of blades. It has repeatedly been experienced that



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the number of maxima and minima, during the above-mentioned period, decreased from four to three, in a vessel with four-bladed propellers, as soon as three-bladed propellers were put on, and *vice versâ*.

If Diagram II. is minutely examined, the following is the result (see Diagram II., Plate XXXVIII.):—

| —— | Horizontal Vibrations. | Port Engine Astern of Starboard Engine. |
|---|---|---|
| | | Degrees. |
| Position A ... ... | Minimum ... ... | 90 |
| Position B ... ... | Maximum ... ... | 130 |
| Position C ... ... | Minimum ... ... | 180 |
| Position D ... ... | Maximum ... ... | 225 |
| Position E ... ... | Minimum ... ... | 260 |
| Position F ... ... | Maximum ... ... | 310 |
| Position G ... ... | Minimum ... ... (is shifted a little) | 360 |
| Position H ... ... | Maximum ... ... | 35 |
| Position J ... ... | Minimum ... ... | 90 |

If one draws the relative positions of the propellers in these different points, it will be found that the propellers are nearly symmetrical for the minima, such as represented in Fig. 12, Plate XXXVI. This figure, therefore, corresponds with the positions A, C, E, G, and J. Since the torsional vibrations are only caused by the propellers, as we have seen, it will easily be perceived from the Figure that no forces whatever can appear which produce torsion; because all act symmetrically, and any forces occurring in the one propeller are destroyed by forces of the same amount in an opposite direction arising from the other propeller.

The case is different when the one propeller leads the other by 45°, that is, when the positions of the propellers are as shown in Fig. 13, Plate XXXVI. In this position there is no symmetry in the forces produced by the propeller blades. According to former explanations, blade A of the starboard propeller will meet with the greatest resistance in the position drawn (about 30°, rising in an outward direction), and, consequently, the shaft will at this moment exert a strong reaction upwards. As the port engine cannot exert the same reaction at this moment on account of its

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position, the after part of the vessel inclines from starboard to port. But, as soon as both propellers have revolved about 45° further, as shown in Fig. 14, Plate XXXVI., the port propeller will exert its greatest reaction in an upward direction, and a force will be produced which will cause the vessel to incline from port to starboard.

As long as the positions of both propellers differ about 45° from each other, as in Positions B, D, F, and H, forces occur which produce torsional or horizontal vibrations, and a vibration maximum is therefore attained; and when the positions of both propellers are nearly symmetrical, horizontal vibrations must disappear, or, in other words, a minimum takes place.

The data already given for the different positions of Diagram II. confirm these facts completely.

In Diagram V., Plate XXXVIII., the maxima and minima for the torsional vibrations may be ascertained in the same manner. In order to understand the diagram correctly, it is necessary to bear in mind that here the number of revolutions is nearly the same as the critical one for horizontal vibrations. The vibrations are, therefore, more violent, and are still perceptible some time after the cause has disappeared. Therefore, the conditions are here considerably different from those in Diagram II. In Diagram V., contrary to what we have found in Diagram II., the minimum takes place when there is a difference of nearly 45° in the positions of the propellers, and a maximum when the propellers are nearly symmetrical. Of course, there will occur a phase-difference in the horizontal vibrations, but they cannot, however, be accurately ascertained on account of the small extent of the amplitude in the diagram.

In this manner the so-called horizontal vibrations, about the causes of which quite anomalous hypotheses have been laid down by others, are easily explained.

Although in the foregoing all important vibrations have been explained in a simple and unconstrained manner, still it will always be of value to bring a further proof to show the correctness of the statements given for the causes of vibrations.

The conclusions I drew from the experiments were, that one blade of the propeller has a greater resistance in turning than the other blades have, or that the opposite blade has a corresponding less resistance.

The correctness of this conclusion enables one to make the following test :—

If my conclusions are correct, *i.e.*, if both the respective blades really offer an essentially greater resistance, a greater force is exerted, which will incline the vessel from starboard to port at that moment when the minimum of vertical vibrations

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takes place, the port propeller being 270° ahead of the starboard one, as shown in Fig. 15, Plate XXXVI.

This is to be explained in the following manner :—

If both the blades are in a symmetrical position when revolving, for instance two blades of each propeller being in a vertical position simultaneously; or when two of the blades form a certain angle to the horizontal position, as shown in Fig. 15, Plate XXXVI.; then, as we have seen, a minimum must take place with regard to the horizontal vibrations. When, however, both the blades A and $A_1$, which have the greatest resistance, are in a position as shown in Fig. 15, Plate XXXVI., there must come a moment when an impulse will incline the vessel from starboard to port; for, according to our former explanations, the blades offer then the greatest pressure in an upward and downward direction. This impulse only acts during a quarter of a revolution, and acts, to a certain degree, in a jerking manner, and does not return till after a full revolution.

Wherever a minimum in the vertical vibrations takes place, a minimum in the horizontal vibrations must also occur; but at every revolution there must appear a small wave in the horizontal vibrations, which corresponds with the above described jerking impulse.

This small wave is to be clearly seen in Diagram II., Plate XXXVIII.: the small waves are indicated by arrows.

After the two propellers have revolved 180° they will be in a position as shown in Fig. 16, Plate XXXVI., when the blades marked A and $A_1$ will again be those which will meet with the greatest resistance. At this moment the vessel will incline from port to starboard. In the same way, as has already been described, a small wave would have to appear at the opposite side, in the curve of horizontal vibrations. This is indeed the case; but, as there are still vibrations caused by the former impulse, this second impulse, which occurs 180° later, can only have the effect of neutralising those vibrations. The result of this is that a curve must appear, which resembles that shown on an enlarged scale in Fig. 17, Plate XXXVI.

The lower curves *c c c*, therefore, correspond with the effect of the blade position as shown in Fig. 15, and the shallow curves *a b* correspond with the blade position as represented in Fig. 16.

The curves for the horizontal vibrations show in Diagram II., Plate XXXVIII., exactly the same form as in Fig. 17, and therefore confirm entirely our conclusions.

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## Concluding Remarks.

If we sum up the results of the foregoing researches we have to record the following:—

The critical number of revolutions for the vertical and horizontal vibrations has been pretty accurately ascertained.

It has been proved that the slight vertical vibrations are neither caused by mistakes in the balancing of the masses, nor by variations in the turning moment, or speed of revolution.

It has been ascertained that the vertical vibrations are caused by a somewhat greater resistance in one blade on each propeller, and it has been shown which these blades are.

The cause of the maxima and minima of vertical vibrations has been found.

It has been proved that each propeller blade finds its greatest resistance in revolving, when outwards at an angle of about 30° to the horizontal position.

It has further been ascertained that the horizontal vibrations are also caused by the propeller blades, and that the number of the maxima and minima which occur during the time that one engine makes exactly one revolution more than the other depends upon the number of blades.

The variations in the turning moments of the engines have no influence on the horizontal vibrations.

Finally, the conclusions we have been enabled to draw from the appearances of the vertical vibrations, viz., that one blade on each propeller meets with a greater resistance when turning than the others, are also confirmed by the curves of the horizontal vibrations.

These results agree on the whole with those which have been found by me on other vessels.

The greater resistance of the one blade, of which we have repeatedly spoken, is probably in most cases to be attributed to the greater pitch of the same. The least deviation in the pitch, which cannot be proved by ordinary instruments, appears to be sufficient to cause perceptible vertical vibrations. The greatest care must therefore be taken in the manufacture of propellers, not only for the sake of preventing vibrations, but also for saving power, and in order to prevent blades from breaking.

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TABLE OF PALLOGRAPHIC RESULTS OF THE STEAMER "DEUTSCHLAND."

| Number of Diagram. | Position. | Revolutions per Minute. | | Advance of Port Engine at Maximum. | Position of Crank (I) when the Steamer is at its mean position. | | REMARKS. |
|---|---|---|---|---|---|---|---|
| | | Port. | Starboard. | | Starboard. | Port. | |
| I. | I. | 62·83 | 62·81 | 55° | 35° | 90° | Very small vertical vibrations. Small horizontal vibrations. |
| | II. | 62·83 | 62·81 | 70° | 20° | 90° | |
| | III. | 62·92 | 62·68 | 70° | − 20° | 50° | |
| | IV. | 62·92 | 62·68 | 100° | 20° | 120° | |
| | V. | 62·92 | 62·68 | 120° | − 30° | 90° | |
| | VI. | 62·92 | 62·68 | 130° | − 50° | 80° | |
| II. | I. | 64 | 65·2 | 100° | − 10° | 90° | Perceptible vertical vibrations. Small horizontal vibrations. |
| | II. | 64 | 65·2 | 80° | + 15° | 95° | |
| | III. | 63·8 | 66·2 | 100° | − 10° | 90° | |
| | IV. | 64 | 66 | 95° | − 15° | 80° | |
| | V. | 64·6 | 66·25 | 100° | 0° | 100° | |
| | VI. | 65 | 66·6 | 90° | 40° | 130° | |
| | VII. | 66 | 66·75 | 100° | 30° | 130° | |
| | VIII. | 66·9 | 67·2 | 90° | 80° | 170° | |
| III. | I. | 67·25 | 67·25 | 115° | 75° | 190° | Perceptible vertical vibrations. Slight horizontal vibrations. |
| | II. | 67·25 | 67·25 | 95° | 85° | 180° | |
| | III. | 67·4 | 67·6 | 150° | 80° | 230° | |
| IV. | I. | 67·3 | 67·75 | 90° | 105° | 195° | Perceptible vertical vibrations. Slight horizontal vibrations. |
| | II. | 67·3 | 67·75 | 150° | 85° | 235° | |
| V. | — | 79·6 | 78·05 | — | — | — | Very small vertical vibrations. Small horizontal vibrations. |
| VI. | I. | 79·47 | 78·82 | — | — | — | Small vertical vibrations. Slight horizontal vibrations. |
| | II. | 79·19 | 79·36 | — | — | — | |
| VII. | — | 66 | 66 | — | — | — | Most violent vertical vibrations. |
| VIII. | — | 75 | 75 | — | — | — | Most violent horizontal vibrations. |

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## DISCUSSION.

Mr. B. Martell (Vice-President): My Lord and Gentlemen, I am sure we all highly appreciate the work which Herr Schlick has done, and our very best thanks are due to him for furnishing this Institution with the results of the investigations which he has made. We all know the great and important results that have accrued from the experiments made by Herr Schlick. They have enabled us to understand the cause of those great vibrations in ships, and he has at the same time brought forward proposals, and carried into effect the means of preventing the unpleasant sensations we have experienced from the vibration of vessels. I think you will all agree that we should express our thanks to Herr Schlick for his continued efforts, and for his great kindness in placing before this Institution the result of his work. It would be very much to be regretted, if we allowed such a paper as this, in which so much work has been accomplished, to pass without conveying to the author our best thanks, particularly seeing that he is away. As you know, gentlemen, Herr Schlick is a foreigner, and I think that we should convey to him our appreciation of his work. We trust that he may be able to continue his valuable investigations for the benefit of all those interested in shipping.

Professor H. S. Hele-Shaw, LL.D., F.R.S. (Associate): My Lord and Gentlemen, my only excuse for making any remarks is that you have requested me to do so, and because I feel that it would be a matter of regret if no discussion took place on this most important paper. If I had known the character of the paper, and had had access to it beforehand, I should have tried to do justice to it in my observations. I may say, however, that I have previously attempted to make some experiments in the same direction. I was first led to think of doing so by Sir Benjamin Baker. I happened to be travelling to America on board the *Teutonic* at the same time that he was, and we both noticed that, in going from Liverpool to Queenstown to pick up the mails, the engines being kept at half speed, there was a vibration of quite a different character to that which took place after leaving Queenstown, when the engines were put at full speed. The *Teutonic*, as we all know, is not a ship that vibrates in a very objectionable manner; but the vibrations are quite marked, as well as their different character. We discussed this subject, and Sir Benjamin spent some time in the engine-room trying, by experiments of an extemporised character, to investigate the matter. Since then I have made several attempts to find out more about the character of the vibrations of a ship, and it is of great interest to see how Herr Schlick has attacked the problem, and to hear the valuable results of his work. I regret very much that we have not the actual instrument to look at, because we could understand better by looking at many little details of importance, but the paper gives sufficient explanation to enable us to appreciate the results which have been arrived at, and which seem to conclusively demonstrate that it is not from the engine itself only that the vibration proceeds, but the screw propellers, and this, I think, will open up a new field of research. I feel that this paper really must mark the beginning of more careful investigations into this question. I feel very grateful personally to Herr Schlick for calling attention to the peculiar action of the screws, and for putting the matter in such a way as will enable other people to follow up his work. I hope this is only the beginning of a series of investigations which will throw more light on the action of the propeller. Now that we are getting to know almost all there is to be known about the dynamical balancing of the engines themselves, it is quite clear that the mutual reaction of the propellers on each other is the next problem which is to be attacked, before we arrive at what engineers are hoping to get ultimately, a perfect balance in a ship without any vibration at all due to the machinery.

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Mr. C. H. Wingfield (Member): My Lord, I regret that I am not able to thoroughly grasp this important paper in the short time it has been in my hands. Herr Schlick's contributions always repay very careful reading. It may be worth while, however, to mention an experience met with by another German company. They had considerable trouble from vibration in a steamer which they had built. The engine was not balanced by Herr Schlick's very complete method. I do not think he had at that time devised it. Therefore there was a certain amount of vibration due to the engine, and a certain amount was also due to the propeller. The propeller had three blades, and by turning it round one-sixth of a revolution the relative phase of the vibration due to the engine and that due to the propeller was altered, and the result was a very considerable improvement. That bears out what the author says about the important effect the propeller sometimes has on the vibration of a ship. I am sure that we all feel a great debt of gratitude to Herr Schlick for bringing this paper before the meeting.

Professor G. H. Bryan (Visitor): This paper is difficult for me to discuss. It appears that when the vibrations of the engines of a ship are overcome by balancing, other vibrations remain which are caused by the propeller. The intensity of these vibrations will necessarily depend on the speed of working, and, should their periods correspond to those of a free vibration of the ship, their intensity may become considerable. It appears, too, from the paper, that the vibrations are altered in character, as one would naturally expect, by increasing the number of blades of the propeller. A further question suggests itself, namely, whether the vibrations could be eliminated by using screw-shaped propellers, instead of ones with narrow blades. I have been extremely interested in the paper, which suggests many possible lines of investigation.

Mr. J. I. Thornycroft, LL.D., F.R.S. (Vice-President): My Lord and Gentlemen, the action of the screw is necessarily oblique; at the same time, from the want of uniformity in the velocity on the various parts of the current following the ship and acted upon by the screw, its blades encounter more or less resistance at different parts of their revolution. This, of course, means that the water is more or less accelerated at different parts of the revolution, and the sum, or resultant, of motion impressed on the water by the propeller makes a considerable angle with the line of the propeller shaft and line of motion of the vessel, if the period of this irregular impulse correspond with any natural vibration of the ship, vibration must result. My firm made some experiments with a particular form of propeller in which the ejected stream is a good deal constrained. I am referring to the turbine propeller. In that you may be aware that the stream is constrained, going out of an annular aperture as it were. It is also guided by a conical body beyond that aperture, so that any irregular stress, or irregular impulse, put on the water, relative to the line of motion, more or less reacts on this body, and the disturbance which I have tried to describe is in that way eliminated, and it becomes self-contained work within the propeller. What we have noticed, I think I may say with confidence, is that in these boats there certainly has been less vibration, considering the light structures and the power put in them. There is another view of the subject to which Professor Bryan has referred, and that is the altering of the number of blades, or changing their form. Practically we want to use as few blades as we can. I think, therefore, we are stopped from having a very great number of blades. If we could—to use a mathematical term—use an infinite number of blades, I take it then the resultant pressure would be in a particular direction, and really we should then have no vibration due to the propeller.

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Professor J. H. Biles (Member of Council): My Lord, I had no intention of saying anything on this paper; but, perhaps, I may be permitted to make a few remarks. The value of this paper seems to me to be not so much in the discovery of the cause of vibration—that the cause of vibration is in the propeller—as the identifying of the particular vibration with the particular propeller which causes it. The investigation is one in which is traced out, I think for the first time, the relation between the observed residuary vibration on deck and the cause of it. We are familiar with the fact that propellers do cause vibration, and especially is that fact familiar to those who happen to have been on board a ship where the propeller has touched the quay wall or a pile in going out of dock. The change in the vibration caused by a very small change in the propeller is quite apparent, so that the fact that propellers do cause vibration is not discovered by Herr Schlick in his paper, and I do not suppose he ever pretended that he did discover it. Therefore, I think that, while we associate with our commendation of this paper the fact that it is a very elegant investigation, if I may say so, still we must keep in mind the fact that the vibrations which were observed were really very small vibrations, and that, even if one could do away with the whole of them, by so improving the propeller as to make it perfectly uniform in its action, yet I do not think it would add very much to the efficiency of the ship. I think that is likely to be borne more forcibly to our minds, if we consider that, although these vibrations are quite perceptible to us in smooth water, and that they are caused by the angular action of the propeller, yet the forces which come into operation in smooth water, and which cause unequal action of the propeller, are small compared with the forces that come into operation when the stern of the vessel is moving up and down in a seaway. Therefore, vibrations eliminated by doing away with this inequality of the propeller are very small compared with the vibrations which will arise, whatever may be done to the propeller, when the vessel is in motion in a seaway. So that, while this is a very elegant investigation, I am not sure that we shall be much disposed to take action in the direction of changing the blades of our propellers, or the number of the blades, or the form of the propeller, in order to eliminate this small final vibration. The work that Herr Schlick has done in removing the great cause of vibration, namely, the want of balance in the engine, is very much greater in my opinion than the work he is doing now, although the paper gives us a most interesting method of finding out the real cause of these final small vibrations.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.):—I am sure it is the sense of everybody present that we have had a most interesting discussion on this paper, and we must all very much regret that Herr Schlick was not here himself to take part in it. Perhaps Professor Busley will be so kind as to let Herr Schlick know how much we regret his absence on this occasion, and thank him for his very valuable paper.

In reply to the discussion on his paper Herr Schlick sent the following remarks:—

Herr Otto Schlick (Member): Having been prevented by illness from being present when my paper was read, I now take the liberty to add a few words with regard to the discussion. As Professor Biles correctly says, it was not my intention to prove by my experiments that vibrations are to a great extent due to the propeller, because this is already a well-known fact; but what I had in view was to show the way in which the propeller creates vibrations, and I think my efforts in this direction have not been without success. My experiments not only prove the importance of correct and truly shaped propeller blades for preventing vibrations in a vertical plane, but also show in which position each blade meets with the greatest resistance, a point which, as far as my knowledge goes, has not



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yet been investigated. These experiments further show that, even with very carefully finished propellers, one of the blades meets with greater resistance than the others, probably on account of a small difference in pitch, and this explains to a great extent the mysterious fractures of propeller blades, for which formerly no explanation could be found. Bearing in mind that the pitch angle, that is, the angle at which the particles of water strike the blades, is always very small, and amounts only to 2° or 3°, a blade with a somewhat too large pitch will have a much greater resistance, and must do nearly all the work. If, on the other hand, one of the blades has a somewhat smaller pitch, it produces considerably less thrust, or no thrust at all, and only increases the friction of the propeller, which, as is well known, is a considerable item. Therefore, I think, that the efficiency of the propeller could be a good deal improved by using properly machined blades to ensure perfect uniformity in the pitch and shape for all blades of the propeller. In answer to a communication by Mr. Rounthwaite, I can only state that the greatest care was taken in finishing the propellers of the *Deutschland*. They were not only properly balanced before leaving the workshops, but each blade was also tested with regard to its proper shape, and was adjusted to the boss with all possible accuracy.

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# THE INFLUENCE OF DEPTH OF IMMERSION ON THE DISTRIBUTION OF PRESSURE OVER A SUBMERGED MOVING PLATE.

By C. Humphrey Wingfield, Esq., Member.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 28, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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In a recent correspondence, an engineer of some eminence put forward the view that the thrust of a submerged propeller was greatest on those parts of the blades which were most deeply immersed, on account of the greater head of water pressing upon them as compared with that over the upper parts of the propeller.

He supported his contention by reference to the fact that it formed the basis of a paper, read before this Society not long ago, by Mr. Scott Younger, and that it was not challenged during the discussion.

Reference to the paper in question shows, however, that it deals exclusively with propellers which are only partly submerged. Of course, in this case, the upper blades will give less thrust than those which are fully immersed.

As it appears that the idea, put forth by the correspondent already mentioned, is by no means confined to one individual, the author has thought a short investigation of the effect of immersion might be of interest.

For the purpose of this paper it is proposed to consider, not an actual propeller, but the more simple case of a plane, moving through water in a direction normal to its own surface, and to examine whether, when moving at any particular speed, it will meet with more resistance (*i.e.*, give greater thrust), if its immersion is increased.

A diagram* has been published, giving a supposititious distribution of pressure on vertical propeller blades, which is made to vary from nothing, at the surface, to a maximum at the lowest point; just as in the case of a tank (Fig. 1, Plate XXXIX.), where the effective pressure per square foot on one side is nothing at A and B C at the bottom, and the centre of pressure is at one-third the depth. In Fig. 2, let A B represent a stationary immersed vertical plane normal to the paper, the upper edge of which

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* Transactions of Institution of Naval Architects, Plate XLIV., Vol. XLII.

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is just awash. Under these circumstances, and provided there is a vacuum above the surface of the water, the diagram of pressures on one side of the plane will be, as in Fig. 1, a triangle A B C. It should not be forgotten however that a similar triangle, A B D, would then represent the distribution of pressure on the other side of the plane, which is therefore *in equilibrio*, so far as these pressures are concerned.

If, as is more usual in actual cases, the surface of the water has to support the weight of the atmosphere (equivalent to a head of about 33 ft. of sea water, or 2,113 lbs. per square foot), the pressure per unit of area will be increased by this amount at each point of the plane, the effect being, as shown in Fig. 3, to move the lines A C and A D, by amounts equal to 2,113 lbs. per square foot, to the new positions $a\,c$ and $a\,d$. These new lines will intersect at a point $a$, which is 33 ft. above the water surface. Obviously, the centre of pressure on one side of an immersed plane (which is at the same depth as the centre of area of the shaded surface in Fig. 3) is by no means at two-thirds the depth of immersion, as it would be if Fig. 2, or Fig. 1, correctly represented the distribution of pressure.

If the plane A B is wholly immersed (Fig. 4), the pressure diagram $a\,c\,d$ will be the same, so long as no motion takes place, as in Fig. 3. Now, let the plane A B move towards the right of Fig. 4, at a uniform velocity of $v$ feet per second; the pressure per square foot, on the forward side, will be increased by nearly $\frac{w\,v^2}{2\,g}$, where $w$ = the weight of a cubic foot of water. This may be conveniently represented by drawing a line $j\,h$ parallel to $a\,d$, and at a distance from it $= \frac{w\,v^2}{2\,g}$.

Similarly, the pressure per unit of area on the following side of the plane is reduced at a rate also varying approximately as $v^2$.

As the water here is greatly disturbed by eddies, however, this reduction will be less than $\frac{w\,v^2}{2\,g}$, and some figures given by Professor Unwin indicate that this negative pressure may be taken as $n\,\frac{w\,v^2}{2\,g}$, where $n$ is, probably, about 0·5 for an isolated plane moving in a large body of water.*

This reduction of pressure may be represented by drawing a line $f\,e$ parallel to $a\,c$, and at a distance from it $= n\,\frac{w\,v^2}{2\,g}$ (see Fig. 4).

The summation of these positive and negative pressures may be conveniently made by drawing through $e$ (Fig. 5) a line $e\,x$ parallel to $a\,d$, and, by construction, at a distance from it $= n\,\frac{w\,v^2}{2\,g}$.

* The values stated for these constants are probably approximately correct. Even if their actual values differ widely from those suggested, the reasoning is unaffected, however.

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The width of the shaded area represents the distribution of effective thrust, which will be seen to be the same at all depths.* The effective thrust per square foot may be written $k \frac{w v^2}{2g}$, where $k = (1 + n) =$ about 1·5. It may be noted that the effect of wave-making is neglected in this investigation, and that the immersion is assumed to be sufficient to avoid drawing air down.

To test the correctness of the conclusion, that the distribution of pressures on either side of an isolated moving plane, due to its motion, is unaffected by depth of immersion at speeds short of that at which cavitation occurs, the following simple experiment was tried (see Fig. 6):—Two open-ended glass tubes, each with one end bent at right angles, were fastened together with their ends pointing in the same direction, and drawn at a steady rate through water in the direction of the arrow.

The respective depths of their bent ends being 2″ and 12″ below the surface of the water, the liquid in A should have risen $\left(\frac{12}{2} =\right)$ 6 times as high as that in B, if the assumption were correct that the thrust on the advancing side of an immersed plane varies as the pressure diagram in Fig. 1.

Both tubes, however, indicated precisely the same increase of pressure.

As a further check, a more delicate instrument was then tried, the construction of which was suggested by a manometer devised by Mr. J. Macfarlane Gray.

The two tubes (see Fig. 7) were, in this case, connected at their upper ends, the bend being filled with oil, and the remaining portions with water. If the density of the water were $x$ times that of the oil, this would give readings magnified in the proportion

$$\left(\frac{1}{1 - \frac{1}{x}}\right) : 1,$$

so that any difference of pressure would be seen as at H.

Here, again, however, there was no movement of the columns; the increase of pressure at A was exactly equal to that at B.

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* At sufficiently high speeds to produce "cavitation," this equality of distribution disappears. In Fig. 5A, for instance, the speed $v$ is so great that the point $e$ is below the upper edge of the plane. The pressure at $e$ on the following surface being reduced, from what it would be were the plane at rest, by an amount equal to the atmospheric pressure plus that due to the head of water, a vacuum is formed here, and at those parts of the following surface of the plane which are above it, and cavitation results. In this case $e$ A forms the boundary of the upper part of the thrust diagram, as shown by the shaded area in Fig. 5A.

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In the case of a propeller, the relative amount of thrust due to the positive and negative pressures is modified by the rotation of the column of water. This action is outside the scope of this paper, which deals only with the effect on the thrust of varying the depth of immersion. So far, only the case of an isolated plane moving in a large mass of still water has been considered. An actual propeller is not isolated, but is in water which has been previously disturbed by the passage of a ship's hull.

This disturbance materially affects the distribution of pressure, since it produces a forward motion of the wake in which the propeller works, the speed of which is different at different depths.

This forward motion is least where the section of the hull is such as to disturb the water least—*i.e.*, near the keel. The forward motion of the water relatively to the ship, and hence relatively to the propeller, being greatest near the surface, it may perhaps be taken for granted that the thrust (which, as has been shown above, would be equal at all depths for equal acceleration,) must be greater near the top where the change of motion is greatest, than lower down;* so long of course as cavitation does not occur and air is not drawn down.

This conclusion is, of course, directly opposed to the proposition referred to at the commencement of this paper.

A further proof of its correctness is furnished by the beautiful experiments of Professor Hele-Shaw.

As a rudder moves in the same direction as the wake, whereas a propeller pushes in the opposite sense, it is self-evident that, if the conclusion that the lower part of a propeller experiences the least resistance in a sternway direction is correct, the corresponding part of a rudder should experience the maximum resistance in a forward direction.

That this is so is seen at once by the concentration of those water lines on Plates XXVII., &c., of Professor Hele-Shaw's paper. (Transactions Inst. N.A., Vol. XLII.), representing the flow near the deepest sections of rudders.

As this concentration indicates a higher velocity than at other sections of the rudder, the pressure due to the increased rush of water against the section will also be greater.

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* That this is the case was the conclusion arrived at by Professor Osborne Reynolds, after a careful study of the influence of propellers on steering. (*Vide* Barnaby's "Marine Propellers," page 55, Fourth Edition.)

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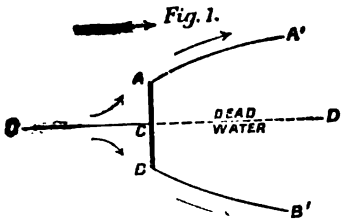
## DISCUSSION.

Professor G. H. Bryan, M.A., F.R.S. (Visitor): My Lord, there is one question I should like to ask Mr. Wingfield. It is not very clear in the paper what way these experiments with the closed tubes were made, because in the reference to the experiments no mention is made of the plane. Was the plate moved through the water with the tubes in front of it, or were the tubes moved through the water without the plate?

Mr. Wingfield: Without the plate.

Professor Bryan: There is one fallacy in Mr. Wingfield's argument on page 68, which may considerably affect the conclusions. Suppose a plane lamina to move uniformly through fluid, or, what would amount to the same thing, a large body of fluid is flowing steadily past a plane lamina. It is true that the effect is to produce an excess of average pressure over the front face as compared with the back face, which is proportional to $\frac{w v^2}{2 g}$, or $\frac{1}{2} \rho v^2$, where $\rho$ is the density; but Mr. Wingfield's investigation is based on the supposition that this excess of pressure is distributed uniformly over the area of the lamina, while in reality this is far from being the case. Consider (Fig. 1) a lamina A B, past which a perfect liquid is flowing steadily in the direction shown by the arrow. Suppose first that all effects of gravity are neglected, and that at the back of the lamina there is a region of dead water, which is separated from the moving part by a surface of discontinuity or free stream lines A A′, B B′. Along these stream lines the velocity in the moving part is constant, and equal, say, to $q$, so that in crossing these lines the velocity changes suddenly from 0 to $q$ without any sudden change taking place in the pressure. The pressure in the dead water is equal to a constant say, P. In the regions of moving liquid, the pressure is determined by the ordinary equation—

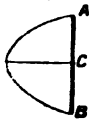
Fig. 1.




$$\frac{p}{\rho} + \frac{1}{2} v^2 = \text{constant.}$$

Consequently, where the velocity of the fluid is less than $q$ the pressure is greater than P. At the centre C of the lamina, where the stream divides, the velocity of the fluid is 0, and it increases from 0 to $q$ in going from C to either of the edges A or B. Hence the pressure on the front faces is greatest at C, being equal to $P + \frac{1}{2} \rho q^2$, and it decreases to P at A or B. The excess of pressure on front over back therefore varies from $\frac{1}{2} \rho q^2$ at the middle down to zero at the edges, and its distribution would be represented by a curve somewhat of the form shown in Fig. 2. We next examine the effect of gravity, on the supposition that the lamina is so far below the surface that the surface conditions do not modify the form of the stream lines. The pressure equation in the moving liquid would then be given by—

Fig. 2.




$$\frac{p}{\rho} + \frac{1}{2} v^2 - g z = \text{constant,}$$

and that in the dead water by—

$$\frac{p}{\rho} - g z = \text{constant,}$$

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if the axis of $z$ be measured vertically downwards. The only effect of gravity would be to increase the pressure in both parts of the liquid by the amount $g\,z$, *i.e.*, to superimpose an added pressure proportional to the depth. As this added pressure is the same on both sides of the lamina, it would not affect the distribution of the difference of pressure on the two sides, and the resultant thrust and position of the centre of pressure will thus be unaffected by gravity, although the distribution of pressure on either face of the lamina considered separately would be altered. The next case considered is that in which the lamina is so near the surface that surface conditions affect the form of the stream lines. We suppose, first, that the lamina C B reaches to the surface, which is covered with a perfectly rigid sheet of ice (Fig. 3). Then the motion exactly reproduces that in the lower half of Fig. 1 cut off by a horizontal plane O C D, and the distribution of pressure is the same as in the lower half of Fig. 2. The pressure curve will be shown in Fig. 4, and the excess of pressure of front over back will decrease from $\frac{1}{2}q^2$ at the surface to zero at B. The actual case in which the surface of the water is free has not been found capable of exact mathematical solution, but some idea may be obtained of what happens from general considerations.

Fig 3.

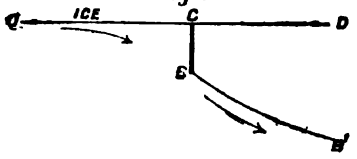
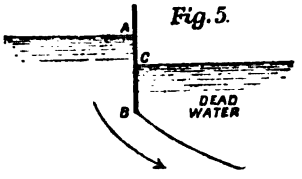



Fig. 4.

Fig. 5.




If (Fig. 5) the lamina projects above the surface, there will be in general a difference of level at the two sides A, C, and this cause alone would produce a difference of statical pressure on the two faces represented graphically in Fig. 6 by the dotted line A P Q (Fig. 6). Since, however, the square of the velocity of the fluid on the front face increases from 0 at A to a maximum at B, the curve representing the difference of pressure when this is taken into account must necessarily be of the form A K B, and the difference of pressure must necessarily either be greatest at C, or attain a maximum above C, and be decreasing from C to B. If the lamina be just immersed (Figs. 7 and 8), with some of the liquid streaming over the top, the distribution of pressure will depend on the position on the plate of the point C, at which the two streams divide, one to flow over and the other under the plate. This will be the point at which the pressure-difference on the two sides will be a maximum. In any case the distribution of pressure over the area of a submerged lamina depends on the form of the stream lines in the neighbourhood of the lamina, and it changes when the lamina is brought so near the surface that the stream lines are modified in form by surface conditions. Mr. Wingfield states that in his experiments with tubes he did not use a submerged plate, but simply moved the tubes through the water. Such an experiments afford no indication of the distribution of pressure on a submerged lamina of finite size. It would be extremely useful if Mr. Wingfield would make experiments with tubes in presence of the submerged plate, when he would certainly observe alterations in the level of the liquid in the tubes indicating the differences of pressure at different points of the plate, and thus determining the distribution of pressure. I am sure that such an investigation would lead to results of considerable interest. While Mr. Wingfield's diagrams fail to correctly represent the "distribution of pressure on a submerged lamina" of finite size (which forms the subject of his title), they would be correct if interpreted as giving a graphic representation of the law according to which the pressures

Fig. 6.

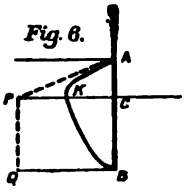



Fig. 7.

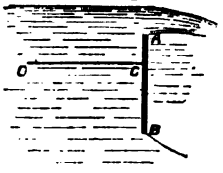
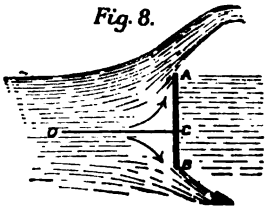



Fig. 8.

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on the two sides of a *very small* moving lamina vary with the depth of immersion. This representation would only fail when the lamina is very near the surface.

Mr. H. M. Rounthwaite (Member): After careful reading, I find the paper unconvincing. The writer of it scarcely goes beyond the statical problem, but implies that his conclusions bear in some way upon the question of propellers, or what might be called the dynamical problem. One might freely admit that planes at rest, or moving at velocities so low that eddies and waves were practically absent, were subject to the laws laid down in the paper, without at all admitting that these laws governed the action of propeller blades. The subject of propellers is difficult, not so much because o the intricacy of its elements taken separately, but because of the complex interaction of the various elements. As to the idea of greater thrusts at greater immersions being confined to one individual, the writer of the paper must know that it was distinctly taught in all the older text-books, and taught, not as a theory, but as a fact determined by experiment. The wide prevalence of the idea might also be inferred from the fact that when, in his paper of last year, Mr. Scott Younger calculated his thrusts on that basis, no one of the two or three hundred members of this Institution present on that occasion seemed to find anything strange about it, or made any objection to the method. The experiments of the late Mr. George Rennie on this subject were made with a two-bladed gun-metal propeller of 21 in. diameter, of the type then used by the Admiralty. He ran the propeller attached to a boat on the river at the uniform speed of 558 revolutions per minute, and found the thrusts vary as follows:—

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Water above tips... | 0″ | 3″ | 6″ | 9″ | 12″ | 18″ | 24″ | 30″ | 36″ | 42″ | 48″ | 54″ | 60″ |
| Thrusts in lbs. ... | 49 | 50 | 196 | 224 | 252 | 280 | 343 | 364 | 369 | 371 | 385 | 399 | 405 |

When plotted, these figures gave a curve of a distinctly parabolic type, excepting that it rose a little at the finish, thus showing a more rapid increase of thrust than in the portion immediately preceding; but this rise did not begin until the water over the tips was more than twice the diameter of the propeller in depth. It will be noticed that when covered by 60 in. of water the propeller gave more than double the thrust noted when it had but 6 in. over it. Professor Osborne Reynolds referred to these experiments and to some similar ones of Maudslay's, in a paper he read before this Institution in 1874, only to condemn them as probably inaccurate, because they did not fit in with the theory he was then enunciating. But it would scarcely appear that such experiments could be put out of court by the mere wave of the professorial hand. Moreover, he believed that no counter experiments had ever been made. No doubt planes at rest, or moving at low velocities, did follow the laws enunciated in the paper, but the blade of a liner's propeller at, say, 2 ft. from the tip, moved at 60 miles an hour. The whole question seemed to depend on the answer to this question:—Does, or does not, the water follow up the blade so as to give the same pressures on the back of it as when the blade is at rest? The answer was given in the paper, and was in the negative. Some figures of Professor Unwin's gave the following pressure as half the pressure on the face. In the discussion on the 1874 paper above referred to, both the late Mr. Froude and Mr. Macfarlane Gray drew attention to the great difference of pressure on the two faces, and to the presence of partially vacuous cavities at the backs of the blades, and Mr. Gray also pointed out that the corrosion which invariably takes place at the backs and near the tips of the blades showed clearly the presence of air, and consequent partial "rupture of column." We heard much of air being drawn down by propeller blades the moment they broke the surface, but it must be remembered that air has no natural tendency to go



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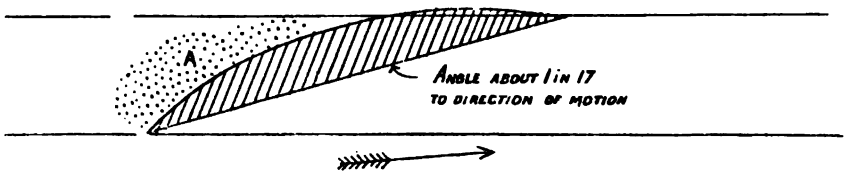
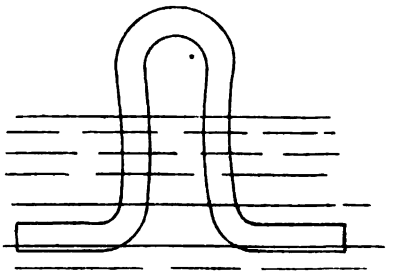
down into water. Therefore, when air goes down it must necessarily be because it finds a partial vacuum into which it can rush. I therefore suggest that the vacuous space, which we all knew as created on first starting a screw, is never altogether absent with any screw, and that, so far as this state of things exists, so far does the distribution of pressures on the blades resemble that of the tank shown at Fig. 1 in the paper. The blade of a liner moves approximately as shown in the sketch (here reproduced), and the corrosion took place just where the eddying and air-charged foam was indicated at A. The glass tube experiment described by the writer of the paper was radically defective, in that it took account of one pressure only, that on the face, whilst the power required to move any plane through water depended upon the algebraical sum of two pressures, those on the face and also on the back. The tubes should have been bent down again as in the figure (annexed), and would then have taken account of both pressures. There are besides many other items of evidence which an engineer or shipbuilder would come across in his daily work, which seem to confirm the view that greater immersions give greater thrusts. The late Mr. Jos. Maudslay had his yacht fitted with a two-bladed feathering screw according to his patent, and when he set the blades in line with the keel and started the engines, the yacht swung round on its centre. He often referred to this experiment as proving that the blades did more work when passing round the lower arc of the circle in which they revolved. Professor Osborne Reynolds, in his British Association Report on the steering of ships, considered that this was due to water thrown against the deadwood and rudder by the blade in passing the top centre; but, as the propeller and hull were rigidly connected, and as the surfaces above and below the centre of propeller were practically of equal area, it appears to me that any force so generated must be entirely external, and could not move the ship. I also think that most of the swinging phenomena which are visible in slowly moving steamers, or steamers just starting from rest, might be accounted for by the same cause, viz., greater thrust at greater immersion. When a vessel is going full speed ahead her gyroscopic inertia is enormous, and the side pressure of the water also so enormous that even a transversely fixed steering screw such as was fitted to the *Hooper* telegraph ship some years ago, was found quite useless. Thus, the irregular action of the propeller has little perceptible effect, 2° or 3° of helm usually covering it. But when the ship is moving slowly or starting from rest the propeller effect is most marked. Then, why in a twin-screw ship like *Iris*, where there is no rudder post behind the screw to carry dead water in front of it, did a two-bladed screw produce such violent vibrations that, though found the most efficient, it could not be used, and had to give way to a four-bladed one, which worked almost without vibration? I suggest that the cause was again the same. There are also many records of cases where three-bladed propellers have vibrated seriously, but six-bladed ones with the same surface ran quietly. Also, Mr. Walker's Bristol experiments (*vide* Trans. of Mech. Engineers) showed similar results. A two-bladed screw gave great vibration and caused the boat to pitch, whereas a four-bladed one was much better, and five and six bladed ones better again, the surface being the same in each case, and revolutions also. The writer of the paper spoke of the following wake currents interfering with the propeller,



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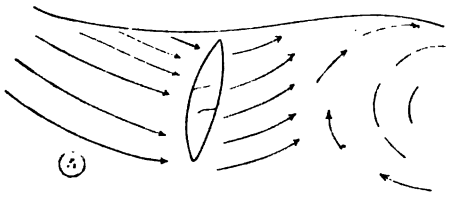
but I suggest that an exactly opposite action takes place, viz., that the screw race neutralises the following wake and reinforces the following wave which is found astern of all ships when moving at speed. The idea is indicated in the sketch (reproduced). It is that the "column of water driven astern" exists only in the imaginations of some writers. Really, the water takes the line of least resistance, and disturbances of surface level take place as indicated. The effect of the following wave is also to cause the lower blades to give a greater thrust than the upper ones. The hill behind the propeller is due in part to the fact that the water driven astern by the propeller cannot go on moving other water to infinity (the revolving annular columns of the text-books), but takes the lines of least resistance, as previously suggested. In the last line of the paper the writer speaks of concentration of stream lines and increased velocity causing greater pressure, but he thought these would clearly cause less pressure. That thrust increases directly as head seems very doubtful. If the following pressure were 50 per cent. of that on the face, then thrust would increase half as fast as head, but our knowledge of the behaviour of eddies as depth increases is very small. It is clear, however, that the same reduction of pressure would cause contained air to boil out of the water under any head, since such air is also under the pressure due to the head. It seems probable that water can be moved by the expenditure of less energy when it is near the surface, and is free to cause surface disturbance, than when its movement is restricted by superincumbent pressure to such paths that a higher velocity of stream line must be generated to effect the translation in the time available—the energy varying as the velocity squared. If this were so, thrust would increase rapidly at first, as the propeller was tried deeper and deeper, and would then increase more and more slowly in proportion as the surface disturbance became less, the thrust curve being just such as the late Mr. Rennie obtained.



Mr. Sydney W. Barnaby (Member of Council): My Lord and Gentlemen, I should like to say a few words on this subject, because I think Mr. Wingfield's paper has hardly received justice. The point he has endeavoured to raise has, I think, been a little missed. Professor Bryan's remarks are very interesting and very thorough, and when we read them in the Transactions, with his diagrams, which I hope will be reproduced, I am quite sure that they will be found to be a most valuable addition to the subject. But I think, really, one can put in a few words what the point in dispute is. Under ordinary conditions of working, when cavitation need not be considered, does the thrust of the propeller vary with the head of water over it? That is, taking a submarine boat, we will say, does the propeller experience more resistance as it goes down deeper into the water, or does it not? Mr. Rounthwaite has alluded to some ancient text-books on the subject. I may say that I have been a student of text-books on screw propellers, and I have found it necessary to discard all text-books more than ten years old. There is one, indeed—I will not call it a text-book, but a book on the screw—printed in America only two or three years ago, where the very view taken by Mr. Rounthwaite was laid down, namely, that the deeper the screw was in the water, the greater the thrust and the greater the resistance it met with. Experiments are adduced in support of this theory, in which the thrust of the screw having no motion through the water was measured at varying depths. Under these conditions air will find its way from the surface to the screw, and affect the result. Doubtless air seldom descends, when a screw vessel attains a good speed through the water, but it does when the vessel is starting from rest, and also in such experiments as those alluded to. There is

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to be found in a book on Resistance and Propulsion of Ships, published in 1898, and written by Professor Durand, of Cornell University, quite a clear statement in accordance with the view which Mr. Wingfield takes, and which I must say I take, that provided the plane or screw is deep enough to avoid surface disturbance, and provided it is deep enough not to draw air, and the velocity is insufficient to produce cavitation, then the resistance or thrust would be independent of the depth, because it is due, not to the pressure, which of course increases with the depth, but to a difference in the distribution of the pressure between the front and rear surfaces. I think Mr. Wingfield has been perhaps a little indiscreet in the title he has given to his paper, because he has rather trespassed on the Professor's playground—"submerged planes." One sees what can be written and said about the resistance of submerged planes; but if we simply take the point which I understand Mr. Wingfield had in his mind, namely the incorrect impression that the propeller of a submarine boat will experience a different resistance at different depths, I think it will be discovered that what he said was sound.

Professor H. S. Hele-Shaw, LL.D., F.R.S. (Associate): My Lord and Gentlemen, the author has stated his problem, or at any rate the point he proposes to clear up, in a perfectly plain and intelligible manner, and he has done service to the Institution by drawing attention to one of the numerous paradoxes which probably abound more in the subject of liquid motion than in any other. He has contented himself with a simple and elementary treatment of the problem, and he is undoubtedly right in the conclusions—at any rate, for certain conditions—at which he has arrived. The title, however, which he has used for his paper, viz., "The Distribution of Pressure," indicates a problem which cannot be treated in an elementary way at all. Professor Bryan has explained to you in a simple and lucid way, and without any unnecessary mathematical symbols, the sort of problem it is. It is one of a most difficult character, although even for a simple flat plate. The problem has never yet been completely solved, although everybody who has gone into the question of the action of the propeller, and the thrust at different points, realises what a vital one it is. Time does not permit me at this stage to say all I should like to say, but there are two points which I do not think should pass without comment. The first refers to the experiments with the two pitot tubes, the action of which Mr. Wingfield illustrates in Fig. 6 of his paper. From the fact that there is the same rise of height in the column of liquid in each of these tubes, independently of the depth of immersion, he infers that the pressure is the same at different depths of a plane surface. But although he undoubtedly records correctly the result of this experiment, I am afraid the result has no direct bearing upon his argument concerning the distribution of pressure upon a plane surface, and merely indicates that the density of a fluid is appreciably the same at any depth, and therefore that an isolated plane surface moving forward, has the same pressure per square inch upon it, no matter at what depth it may be, providing its velocity is the same. If, however, Mr. Wingfield had taken an inclined plate, and by boring holes, distributed his tubes in different positions over a plane surface, he would have found that the pressure was no longer the same in the tubes except at the top and bottom of the plate, but that at intermediate points it would have varied in an appreciable manner, and he would, in fact, have arrived approximately at the results indicated on theoretical grounds by Professor Bryan. The second point which I should like to call attention to is the fact that a screw propeller blade is an inclined surface, and not a plate moving forward with its plane normal to its direction of motion. In some propellers the tips are inclined forward, and some have been tried with the tips inclined backwards. Now in these cases it can be proved that the pressure even on the top and bottom of the plane surface is no longer the same, and, as Mr.

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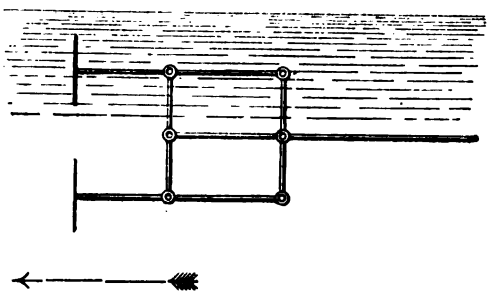
Wingfield has been good enough to refer to one of my papers, I would point out that Plate XXV., Vol. XLII., Proceedings of Institution of Naval Architects, shows at a glance that an inclined plane moving forward with the lower edge in advance would give a maximum pressure near (although not exactly at) the bottom edge, but that, if the inclined plate be moved forward with the upper edge in advance, the maximum pressure would occur near (although not exactly at) the upper edge; in short, the maximum pressure would occur near the edge of a plate which was in advance in either case, so that whether it was greater at the top or bottom would depend entirely upon the mode of inclining the plate. Thus, whereas for a plane surface moved forward in a direction normal to the stream, although it is true, as Mr. Wingfield says, that the pressure is equal at the top and bottom of the plate, it may be seen from Fig. 16 in a Friday evening discourse on "The Motion of a Perfect Fluid," reprinted in the Proceedings of the Royal Institution of Great Britain for 1899, that between those two points the pressure varies continuously, and is a maximum at the centre of the plate, the variation being symmetrically distributed on either side of the centre.

Mr. C. Humphrey Wingfield (Member): My Lord and Gentlemen, Mr. Rounthwaite has designated my paper unconvincing. I think that, if that is so, the discussion has filled up any deficiencies in this respect, and that we must all be especially grateful to Professors Bryan and Hele-Shaw for their contributions. I gather that, with the exception of Mr. Rounthwaite, all the speakers agree that I have proved my point—viz., that the lower blade of an immersed propeller does not experience more resistance than the upper one. Mr. Barnaby referred to submarine boats, and it occurred to me, while he was speaking, that, if it were true that the thrust of a propeller increased with the immersion, there would be a depth at which no material could be got to stand the endwise pressure on a screw shaft, since the hull must also encounter greater resistances with greater depth. Happily that is not found to be the case. Mr. Barnaby has already dealt with the question of air being carried down. I have often seen it coming up (particularly when going astern), and it must first have been drawn down. Several speakers alluded to my experiment with the tubes, and suggested it was inconclusive, owing to the absence of the plane. Since the paper was written I have experimented with a wooden framework similar to a letter balance. Holding this so that the two scale pans were in one vertical plane, and moving it through water in the direction of the arrow, I found that the effective thrust* was the same on both, unless the upper plane was nearly awash. When allowed to disturb the surface greatly, the resistance of this plane became less than that of the other, but not otherwise; and it may be of interest to know that I was unable to detect any difference produced by placing a horizontal glass plate under water and about $\frac{3}{4}$ in. above the top edge of the upper plane.† The action of the two-bladed propeller of infinite pitch, referred to by Mr. Rounthwaite, in driving the stern round in a direction showing that the



* I tried to measure the thrust, but, as my measuring apparatus was limited to 12 oz. (or 6 oz. per plate), I can only say that it greatly exceeded this amount. The balance easily turned with a weight of two grains.

† The plates of the balance shown in the figure measure $2\frac{1}{4}$ inches vertically by $4\frac{1}{8}$ inches horizontally, their centres being $7\frac{1}{2}$ inches apart.

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lower blade was most effective, was due to the fact that, as Mr. Rounthwaite had himself mentioned, there were reactions, in opposite senses, of the water against the upper blade and against the hull or sternpost. These cancelled each other to some extent, and the lower blade was therefore able to produce a greater effect sideways, notwithstanding the greater resistance of the water to the motion of the upper blades. Mr. Schlick's paper, read this morning, confirmed, if confirmation were needed, the point I sought to prove in the paper. It only remains for me to thank the meeting for the kind way in which my contribution has been received.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): I am sure you will all join in a hearty vote of thanks to Mr. Wingfield for the admirable and suggestive paper which has led to so interesting a discussion. We also owe our thanks to Professor Bryan for his most lucid explanation of a somewhat abstruse problem.

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# ON A NEW ASSISTANT CYLINDER.

By David Joy, Esq., Member.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 28, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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I find that it is ten years since I last had the honour to read a paper before this Institution, on the results of my work in that particular branch of marine engineering which has of late years been engaging my attention.

Ten years' steady work in a given direction means a good deal of valuable experience, and should produce a series of solid improvements and advances in constructive details; and it is the resulting improvements and advances, both in construction and efficiency, that I propose to lay before you this evening, with such illustrations as, I hope, will make them both clear and interesting.

The special machinery, originally described under the name of the "Steam Assistant Cylinder," for valve gears, which, I submit, sufficiently indicated its functions, has, I think, pretty fully justified its existence, by the fairly wide adoption it has met with during the last eleven years, by its application on a good deal over 1,200,000 I.H.P. of steam machinery, both naval and commercial, and on both large and fast-running engines, as well as small and slow-running ones.

The early machines* (illustrated by the two first ever built, the complete one and the sectioned one on the table) were fitted on H.M.S. *Speedwell*, in 1889, and were removed a month or two ago when the ship was re-engined, after about eleven years. These were not free from complication, as instanced by the valve for the distribution of steam to the cylinders, and its elaborate adjustments; but these were necessary, as the valve received its movement from the pressure of steam alternately from each end of the cylinder. This complication soon disappeared, and the piston itself became its own valve. But that change necessitated a longer piston, to accommodate the ports and the necessary travel, so a long cylinder followed. Though the machine was larger, it was much simpler, and more efficient; but the ports, being in the piston and in the

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* Transactions Institution of Naval Architects, Vol. XXXII., Plates XXXIII., XXXIV., and XXXV.

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sides of the cylinder, exposed, relatively, larger surfaces than was expedient to pass leakage.

Meanwhile, the indicator cards, taken constantly and studied carefully, were forcing on lessons suggestive of refinement and economy, and thus the modern and approved practice of main engine cylinders was followed. The aim at compounding soon ensued, and suggestions on all hands arose of improved constructive details, so that the remodelling of the whole machine became unavoidably necessary.

It is the newer and improved form that is illustrated by the diagrams on the walls, and by the movable models on the table before you.

This machine includes, as its first important feature, the compound treatment of the steam, which will be explained in due course.

The next step was the transference of the steam ports from the body of the piston and the sides of the cylinder, where so much surface is necessarily exposed to leakage, to a ring round the valve rod in the gland, thus making the valve rod effectively the valve, and reducing very largely the surfaces exposed to leakage. The result was a considerable decrease in the size and weight of the original machine, making it available in many cramped positions, such as under confined armour decks, where before its height rendered it prohibitory. These changes simplified the action and adjustments throughout, so that it now requires fewer adjustments, and the engineer is relieved from the confusion they often induce. Improvement has been made in many other minor details, such as often follow any general change, especially if that change is in the direction of simplification.

We have, now, therefore, a more refined machine, using steam "Compound," and so up to date, yet smaller, lighter, and simpler, and also with a gain in economy.

Now, turning to the diagram model on the wall, I will explain the general structure of the new machine, and show how the various improvements aimed at have first been suggested, and then attained. Of course, as was anticipated, the indicator cards, taken from the first machines gave very much fuller diagrams for the upward, or lifting, stroke than for the falling stroke. But this was so strikingly the case as to suggest that, by a suitable arrangement, the steam might be manipulated in such a way that part of it, after cushioning the downward stroke, and then lifting the piston and its accessories, might be made to provide also for the upward cushion and the falling stroke, and so do duty over again, though at a reduced pressure. In this way the advantage of compounding is secured. It will be seen how this comes about by following one complete stroke of the piston, commencing at the middle of the stroke, downwards. If we analyse, even superficially, what must be the effects of the admission and distribution of the steam, at the various positions, under the various conditions, we shall see that, with the lowering of the pressure at each following transfer, a less amount of

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work is demanded from the steam, and so an easier and softer action in cushioning and driving in the falling stroke is obtained from it. An almost automatic government ensues, suiting exactly the requirements the machine was designed to effect.

The piston rod, now arranged to act as the valve for the lifting stroke, by the reduction of its diameter passes through the steam space surrounding it formed in the gland, and admits into the bottom of the cylinder a charge of steam to act first as cushion for the downward stroke, and then as the lifting power for the next upward stroke. Also the piston, which is made hollow and acts as a reservoir, is provided with small ports on each side (to keep the piston in balance) bottom and top, as shown. These ports A and B pass over recesses in the cylinder sides C and D, and these complete the portways. Thus, as the piston rises, there passes a charge of steam from under it into the inside of the piston, of course at a reduced pressure. This steam is carried up with the piston till the suitable point is reached, when the upper ports B, with the recesses D, complete the communication from the inside of the piston to the top of the cylinder, filling that with a softer charge of steam to suit the cushion for the lifting stroke. Here the steam is shut in by the further rise of the piston, and, on the return stroke, this steam serves as the assistant downward power, till released by the passage of the piston's upper edge over the exhaust ports in each side of the cylinder, so completing the cycle of the full up and down stroke.

Thus is provided the admission of steam, worked on the compound principle, for both the bottom and top of the cylinder. Also the required cushioning for both top and bottom, as well as the final release, are provided for by the movements of the piston and piston rod, without the addition of any new or extra parts or moving levers, &c., to shake and get loose. In every position all the moving parts are kept in perfect balance by the pressure of the steam surrounding them, and therefore are not subject to any steam pressure causing friction.

It now only remains to add that, after the first experimental cylinder had been made and tested by my firm on one of the steamers of the General Steam Navigation Company (the S.S. *Bittern*), where, by the courtesy of their engineer manager, Mr. J. Preston, we received every convenience for testing the machine to the fullest, the plan was taken up by Messrs. Laird, of Birkenhead, by whom the original type of these assistant cylinders was first introduced on the Government torpedo destroyers *Speedwell* and *Skipjack*. This newer and improved type was also applied to the designs of their general engineer manager, Mr. R. R. Bevis, who introduced it on two steamers with very satisfactory results. In the application of this new type, as in the earlier examples on the destroyers, we have had the advantage of his great practical experience and kindly assistance.



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The following figures illustrate the paper:—

Fig. 1 (Plate XL.).—The first new type cylinder, as fitted on the General Steam Navigation Company's S.S. *Bittern*.

Fig. 2 (Plate XL.).—The first four cylinders built by Messrs. Laird.

Fig. 3 (Plate XL.).—Cylinder fitted on Messrs. Bailey & Leetham's S.S. *Sultana*.

Fig. 4 (Plate XLI.).—Indicator Diagrams from the three cylinders illustrated.

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## DISCUSSION.

Mr. F. W. Crohn (Visitor): I should like to ask Mr. Joy what proportion of the total indicated horse-power of a marine engine is obtained by his assistant valve gear. I mean to say, the work done by this assistant machinery. I should like to know how much work that does in proportion to the total indicated horse-power of any marine engine.

Mr. Basil Joy: I am afraid I shall have to trust to memory for figures there, but, to the best of my recollection, some cards on which we worked out the horse-power were from H.M.S. *Glory*. I think she was 18,000 H.P.—I am not quite clear whether it was 18,000 H.P. or 16,000 H.P.—I am told 16,000 H.P., and the indicated horse-power of these cards was 12 H.P. from the low-pressure cylinder.

Mr. Crohn: Each engine?

Mr. Joy: Each cylinder.

Mr. Crohn: With two low-pressure cylinders?

Mr. Joy: Yes, twin screw.

Mr. Crohn: That would be 24 H.P.

Mr. Joy: Yes.

Mr. F. Edwards (Member) sent the following communication on Mr. Joy's paper: It is convenient for many reasons in a set of triple engines to have the valve motion of all three exactly the same size. The power required to drive each of the three valves is, I believe, invariably different, and the particular valve which requires the most power frequently leads to much trouble in connection with its valve gear. To put the matter right under the present system of designing these engines, all three valve motions frequently require materially strengthening in all the details, or, if this is not needed, some of these gears have been made unnecessarily strong; it therefore appears to me that the use of Joy's assistant cylinder does not necessarily increase the total weight, but renders a reduction possible. One of the first duties of a designer is to transmit his loads, and particularly alternating loads, from one point to another as directly as possible, and, if practicable, to avoid passing through working parts. In one ordinary valve motion there are no less than eleven working

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parts brought into use in transmitting the power from the crank shaft to the valve, and in this large number I have not included either the stern gear or the reversing shaft bearings. In the last engine to which I have fitted Joy's assistant cylinder, which has its valve placed at the back of the cylinder, there are no less than fifteen working parts, exclusive of the stern gear, &c. In both the preceding cases many of these working parts require very careful adjusting, because the rods of which they form part are in pairs, and must therefore be exactly the same length. It is, I think, somewhat remarkable that the ordinary well-designed link motion should have established as satisfactory a reputation as it has, because I imagine that the very smooth link bars with an equally smooth sliding block on them, both well oiled, are, per se, about the worst possible combination for transmitting the load in the desired direction when the links are at their extreme angle. It is true Mr. Joy's method of driving the valve introduces a new part, and that the introduction of a new part is undesirable; but, on the other hand, it is very simple, and practically does the whole of the work, and such small cylinders give very little trouble, and, when requiring repairs, can be easily carried ashore and put in the lathe. The most striking feature in Mr. Joy's method is that the working load is transmitted direct without passing through one working part, and thereby reducing the load, repairs, and oil, on at least eleven working parts. I am aware that these remarks will not carry weight with some of my friends, but I know they will be well received by those who have had trouble with valves, say, 5 ft. square, with a considerable pressure on the backs of them. I have fitted several assistant cylinders with most satisfactory results; in one set of small high-speed, noisy engines, it was not possible to alter the valve gear, and, as a last resource, I fitted one of these cylinders, and was amply rewarded.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): We have heard this interesting paper read by Mr. Joy, Jun., in place of his father, who, unfortunately, is unable to be present to-day. It does not appear from the silence with which it has been received that it is a paper which lends itself very much to discussion. I think you will join with me in thanking Mr. Joy for his paper, and in asking his son to express to him our great regret that we have not had the pleasure of seeing him here to-day to read his own paper.

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# THE GEOMETRY OF ENGINE BALANCING.

By J. Macfarlane Gray, Esq., Member of Council.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 28, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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Engine balancing is the subject of papers in our Transactions by Messrs. Yarrow, Schlick, Mallock, Sankey, Dalby, and myself; its principles are, therefore, fully explained, and on this occasion only drawing-office methods of applying these principles will be considered.

In many engine works the balancing problem has already become an established part of the regular work of the draughtsman, and is no more thought of than settling the lap and lead and angles of the eccentrics for the required cut-off used to be, and with this great experience methods better than those now to be described may have been perfected. In the discussion some of these may be brought forward.

In comparing the balancing of different engines, it is necessary to eliminate the element of mere size, and to express the results obtained according to some intelligible standard of unification. The mean piston-line reciprocating weight is in this paper represented as 1,000. Half the distance between the centres of the extreme cylinders is taken to be the unit tilting arm. The piston stroke is unit travel. For example, if the mean piston-line reciprocating weight is 1,540, and the reciprocating weight of a slide-valve-line is 740, and the travel of the valve is one-fourth of the travel of the piston, then $740 \div 4 = 185$ is the actual equivalent reciprocating valve weight, and $185 \div 1{,}540 = 120$ is the unified valve-line reciprocating weight to be used in the balancing problem. If the centres of the extreme cylinders are 120 in. apart, 60 in. is the unit tilting arm; and if a slide valve centre is 75 in. from the central plane, $75 \div 60 = 1{\cdot}25$ is the arm by which the unified weight 120 is to be multiplied for the tilting moment in the problem.

If a tilting moment is in this way calculated to be 670 for any engine, this will mean that the inertia force of 67 per cent. of the mean reciprocating piston-line weight, acting at the extreme cylinder, is the measure of the tilting moment. According to this system of unification, the comparison numbers will be the same whether the

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calculation be made on the metrical system or in tons, pounds, feet, and inches. By making the mean weight 1,000 instead of unity, we get rid of the decimal point in the comparison numbers.

Only the unified weights are employed in this paper; the actual weights are suppressed, as I have not been authorised to publish the data kindly furnished by makers for my own use, and they are unnecessary for the understanding of the methods.

The scheme of lettering adopted is, a letter A, B, C, D, &c., for each cylinder with an even number, 0, 2, 4, &c., to distinguish the parts, for weights, and the same letter with the next number, odd, for the tilting moment of that weight. Example: for parts in connection with cylinder A we have A, the piston-line reciprocating weight; A 1, the moment of A; A 2, the crank-revolving weights; A 3, the moment of A 2; A 4, the valve-line reciprocating weight; A 5, moment of A 4; A 6, the valve-line revolving weight; A 7, moment of A 6; A 8, the balance-weight at A, if any; A 9, moment of A 8; 10, 12, other balance weights, if any; 11, 13, moments of 10, 12; and similarly for B, C, D, &c. The signification of these numbers ought to be kept in mind. Preparatorily to making the balancing diagrams, the actual weights and arms are tabulated, and from that with the slide rule in hand the unified weights and arms are written out in a similar table as follows. The column 0 refers to the piston-line weights; the 0 is not written after the letter unless when its omission would lead to doubt.

ENGINE NO. 1.—FIGS. 1 TO 4.

| | Cyl. Arms. | 0 | 1 | 2 | 3 | Valve Arms. | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| A | 1· | 985 | 985 | 870 | 870 | 1·365 | 130 | 178 | 159 | 217· | 352 | 352 |
| B | ·515 | 1,005 | 517 | 870 | 449 | ·142 | 136 | 19 | 159 | 22·6 | | |
| C | ·515 | 1,005 | 517 | 870 | 449 | ·142 | 136 | 19 | 159 | 22·6 | | |
| D | 1· | 1,005 | 1005 | 870 | 870 | 1·365 | 136 | 185 | 159 | 217· | 352 | 352 |

The cranks are at right angles, as shown in Fig. 1. The cylinders and valves are arranged as in Fig. 2. There is an air-pump on A reducing the effective weight from 1,005 to 985. The cranks and angles of eccentrics are shown in Fig. 3, looking aft. The direction of rotation is from left to right. In Fig. 2 the central plane is drawn midway between the extreme cylinders A and D, and the distance A X = X D is entered in the column of cylinder arms as unity. Similarly, B X and X C are entered each as ·515, but these are not always alike. In the valve arms column the

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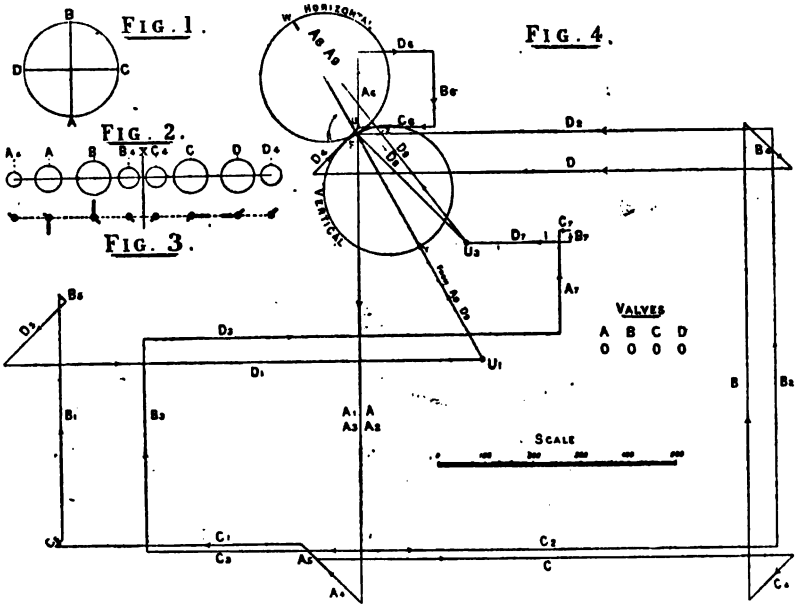
distances from X to A 4, B 4, C 4, and D 4 are entered. The actual weights of the piston-line have been added together, and the sum divided by 4 was, say, equal to 1000 M. The actual weights were then each divided by M, and the quotients, or unified weights, 985, 1,005, &c., entered in column 0. In column 1 are written the moments of column 0, obtained by multiplying each weight by the arm preceding it on the same line. In column 2 are entered the unified revolving-crank weights obtained by dividing the actual equivalent weights, as at crank radius, by the same M. Column 3 gives the moments of column 2. The other columns give similarly the valve arms, the unified valve-line reciprocating weights in column 4, moments of these in column 5, valve-gear revolving weights consisting of the sheaves and the eccentric straps in




column 6, moments of these in column 7. Balance weights at A and D are entered in column 8, and their moments in column 9.

Having completed the tabulation, the diagram is drawn as in Fig. 4, beginning from the point F. From columns 0 and 4 the lengths A, A 4, C, C 4, B, B 4, D, D 4, to U, the ultimate point, are drawn, each in the direction of its centre line for the position chosen to be represented in the diagram. As these lines can be drawn in any order of sequence it is well to make a lettered hand sketch first of how the lines will fall together. Until the draughtsman gets into the way of it he is very apt to draw the eccentric lines erroneously. Instructions are useless to prevent this; he must think how the eccentric radius leads the crank at the opposite side of the shaft with an outside slide valve, and follows it at the same side of the shaft with an inside slide valve. By outside and inside is here meant a valve to which the steam is

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supplied outside or inside. The examples given can be drawn over again for experience and verification. The valves of No. 1 are all outside valves, that is, the steam is admitted at the ends and the exhaust is to the inside of the valves. It must be remembered that the direction in which the lines are drawn is *from* the centre of the shaft. Spend time in verifying the weights diagram before going on with the moments diagram, then the directions have to be the same in the moments diagram for A and B lines, and the opposites for C and D lines. The ahead gear alone is taken in column 4, and the centre of the eccentric is placed at 45° and ·9 of the actual radius; this allows for the weight of astern gear. The lines for all the diagrams start from the point F, and the diagram should finish at the same point. There are actually four diagrams in the figure, the A 0 or A, the A 1, the A 2, and the A 3 polygons. The A 2 polygon, having its sides A 2, C 2, B 2, D 2, and A 6, B 6, C 6, D 6, closes. The others do not close, and their ultimate points are lettered U, U 1, U 3. The scheme of lettering and the darts on the lines make it easy to follow the diagram. Here F U may be disregarded, it is so minute. In this example A 6 starts from U instead of from F to make the figure clearer; as this portion is in itself a closed polygon this is allowable.

The columns 0 and 4 make one polygon, that of the reciprocating weights. The columns 2 and 6 make the polygon of revolving weights. The columns 1 and 3 make the polygon of reciprocating moments, and the columns 3 and 7 the polygon of revolving moments. Having completed the laying off, we have the following resultants:—

F U1 = 540, a reciprocating moment.

F U3 = 320, a revolving moment.

These are cumulative, and together represent a vertical reciprocating moment 850, and a horizontal reciprocating moment 320. In this case these were reduced by the two balance weights in column 8 to a residual disturbance equal to one revolving moment 270. The construction for this is as follows:—Join F U1, bisect in Y, produce U1 F, and make F W = Y F. Join W U3 and bisect in Z. Mark Z W as A 8 and A 9. Mark Z U3 as D 8 and U3 Z as D 9. The balance weights are Z W at A and Z U3 at D. The weight A 8 is to be placed in the direction Z W in relation to the direction of the crank A in the diagram. A being on the positive side of the central plane, the moment A 9 is in the same direction as A 8. The equal weight D 8 is to be placed in the direction Z U3; this is indicated by the arrow before D 8. As D is at the negative side of the central plane, its moment D 9 is in direction opposite to the weight. The moment D 9 is, therefore, in the same direction as A 9, and the two compose the moment U3 W.

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It is instructive to examine this in detail. Green is popularly regarded as composed of blue and yellow. If, therefore, we agree to denote rotational motion by green, we can distinguish vertical motion by blue, and horizontal motion by yellow, and the nature of the forces and moments can then be kept before the eye by this analogy. Draw the line U3 W in green, it is the moment of the revolving weights A 8 and D 8. But U3 W is equivalent to U3 F + F W. The U3 F closes the 3 polygon, and there are left the green F W and the blue F U1, or a blue and a yellow F W, and, in the opposite direction, two F W in blue. A blue cancels an opposite blue and leaves a yellow F W and an equal blue F Y in the opposite direction. These make up a green residual disturbing moment whose magnitude is represented by yellow F W and blue F Y. This resultant green revolves *reversely* to the motion of the engine, because the direction F W is opposite to the direction F Y, and, therefore, it cannot be annulled by a balance weight moving with the engine. The reciprocating moment 540, and the revolving moment 320, have been together reduced to a revolving moment 270 at arm unity.

The fundamental assumption on which balancing methods are based is, that the motions considered are either circular or else rectilineal components of circular motion, as the travel of a piston is, when the crank has uniform velocity and the crank pin slides in a slot connecting rod. Any number of such motions of different magnitudes, having the same period are, in balancing, equivalent to one similar motion of the same period, circular or else reciprocating, and our problem is to find the magnitude and the direction of this resultant at the instant for which the diagram is constructed. In Fig. 4 the resultants are F W yellow, and F Y blue. In the moving diagram now exhibited, circles have been described upon these arms, and the discs are cut out to expose the stationary vertical and horizontal lines, blue and yellow. The circles are mounted on an axis, and revolved as the engine revolves; the components of a revolving weight represented by these arms are then seen, the horizontal yellow intercept, and the vertical blue intercept. As the yellow circle refers to a horizontal reciprocating weight only, we must take notice of only the yellow line through the yellow circle, and similarly we must take note of only the blue line through the blue circle. On another axis there is mounted a single green circle of equal diameter also with the blue and yellow stationary lines. The reciprocating moment-resultants in Fig. 4 are together equivalent in magnitude to the moment of one equal revolving weight, or, the green circle should give the same components as these two circles, blue and yellow. I have mounted these before you to show you that they are of equal magnitude at equal angles, but the yellow in the green circle is always opposed to the yellow in the yellow circle, when revolved in the same direction as the engine. If, however, we revolve the green circle in the opposite direction, the components are the same, and in the same directions, at equal angles from the vertical, and this represents the effect of the

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addition of the balance weights A 8 and D 8 in Fig. 4. This can be shown in another way: writing G, B, Y, for green, blue, and yellow, and resolving the G, B, and Y resultants into their blue and yellow components at angle $a$, and at $(a + 180°)$, and at angle $(-a)$, we get

$$\begin{cases} \text{B at } a & \text{gives} \quad B = \cos a, \\ \text{Y at } (a + 180°) & \text{gives} \quad Y = -\sin a, \end{cases}$$

$$\text{G at } (-a) \quad \text{gives} \quad \begin{cases} B = \cos(-a), \\ Y = -\sin a. \end{cases}$$

But $\cos(-a) = \cos a$, therefore

$$\text{G at } (-a) \quad \text{gives} \quad \begin{cases} B = \cos a, \\ Y = -\sin a; \end{cases}$$

or the same as

$$\begin{cases} \text{B at } a, \\ \text{Y at } (a + 180°). \end{cases}$$

The secondary, or obliquity effect is easily calculated in this example. The elements are tabulated as follows by inspection of the diagram:—

No. 1.—Obliquity Effect.

| | A | B | C | D | Sum. | ÷ 4 | Hyp. | Virtually. |
|---|---|---|---|---|---|---|---|---|
| Cos 2 | 985 | 1,005 | − 1,005 | − 1,005 | − 20 | − 5 | 5 | 1 |
| Sin 2 | 0 | 0 | 0 | 0 | 0 | 0 | | |
| | A 1 | B 1 | C 1 | D 1 | | | | |
| Cos¹ 2 | 985 | 518 | 518 | 1,005 | 3,026 | 756 | 756 | 189 |
| Sin¹ 2 | 0 | 0 | 0 | 0 | 0 | 0 | | |

To be comparable with the numerical primary resultant these obliquity results must be divided by 4. These forces act relatively for only half the time in each impulse, and just as a falling body falls a quarter of the height in half the time, so these obliquity effects have to be similarly reduced.

The irrational expression Cos 2 is a contraction for A Cos 2 $a$, &c., and so with the other lines. Here $a$ is the angle from the upright vertical to the centre line of Crank A. The angle $a$ measures 180°, twice that is 360°, or 0°, and its cosine = 1, and A = 985; therefore, A cos 2 $a$ = 985. The angle $b$ is = 0, and twice 0 is still 0, and the cosine is therefore, again = 1, and B = 1,005; therefore B cos 2 $b$ = 1,005. The angle $c$ = 270°, and twice 270 is 540, or 360 + 180, and the cosine of 180° is − 1, and C = 1,005; therefore C cos 2 $c$ = − 1,005. The angle $d$ = 90°, and twice 90 is 180, and the cosine of



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180° is − 1, and D = 1,005; therefore D cos 2 $d$ = − 1,005. The other lines are taken in the same way. The hypotenuse of the two sides in lines cos 2 and sin 2 is, therefore, = 5, which is the maximum obliquity lifting effect, a negligible quantity, being only the half of one per cent. of the piston-line-reciprocating weight. This 5 would be the radius of the rosette for obliquity lifting, if it were drawn as on p. 162 of Transactions for last year.

The obliquity tilting effect is in this case very great, 76 per cent. of the tilting effect of the mean piston-line-reciprocating weight at one of the extreme cylinders. This is, however, again, only a see-saw, it is not a lifting force. The radius of rosette for this would be 756.

These unified expressions are not in tons, nor in foot-tons. The 756 above means that, expressing the effect of the mean piston-line-reciprocating weight in the position of one of the extreme cylinders as 1,000 at any speed of the engine, this obliquity effect would be 756 in intensity.

No. 2, represented by Figs. 5 to 10, is a large, long-stroke engine, with its cranks also at 90°, but not in the same sequence as in No. 1. The reciprocating weights are dealt with in Fig. 5, and the revolving weights in Fig. 6. Figs. 7 and 8 are enlargements of crowded portions of Fig. 6. Fig. 9 shows the arrangement of cylinders and valves, and Fig. 10 shows the angles of cranks and eccentrics. The unified tabulated weights are as follows :—

ENGINE NO. 2.—FIGS. 5 TO 10.

| | Cyl. Arms | 0 | 1 | 2 | 3 | Valve Arms. | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|---|---|
| A | 1· | 876 | 876 | 1,190 | 1,190 | 1.29 | 49·8 | 64·2 | 46·0 | 60·0 |
| B | ·388 | 910 | 354 | 1,190 | 461 | ·70 | 74·5 | 52·1 | 46·0 | 31·5 |
| C | ·29 | 1,205 | 349 | 1,190 | 344 | ·113 | 104·0 | 11·7 | 46·0 | 4·4 |
| D | 1· | 1,006 | 1,006 | 1,190 | 1,190 | ·69 | 85·4 | 59·0 | 46·0 | 32·5 |

This example is interesting, as I am told that the arrangement of cranks is similar to that of the *Terrible*, originally; the *Powerful*, resembling No. 1. In Fig. 5 the residual vertical force arising from the reciprocating weights is represented by the vector circle on F U = 165. The revolving weights in Fig. 6 have a force resultant, the vector circle which could be drawn on F U2 = 135 omitted for clearness. The maximum of the two combined is = 65, or 6½ per cent. of the effect of the mean

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piston-line-reciprocating weight. The longitudinal tilting moment, due to reciprocating weights, is represented by the vector circle on F U1 = 2,090, and that due to the revolving weights in Fig. 6, by the circle on F U3 = 2,500. Combining these gives the vector circle of total longitudinal tilting with its diameter = 4,660, not shown. This represents an enormous disturbance, exceeding the effect of placing the cylinders two and two in the extreme positions, each pair on one crank, and the two cranks at 180°, which would give a resultant circle only 4,000 in unified measure. It is well that this is only a see-saw result; were it a lifting resultant, it would be intolerable. There are no balance weights in this example.

FIG. 5.

FIG. 6.

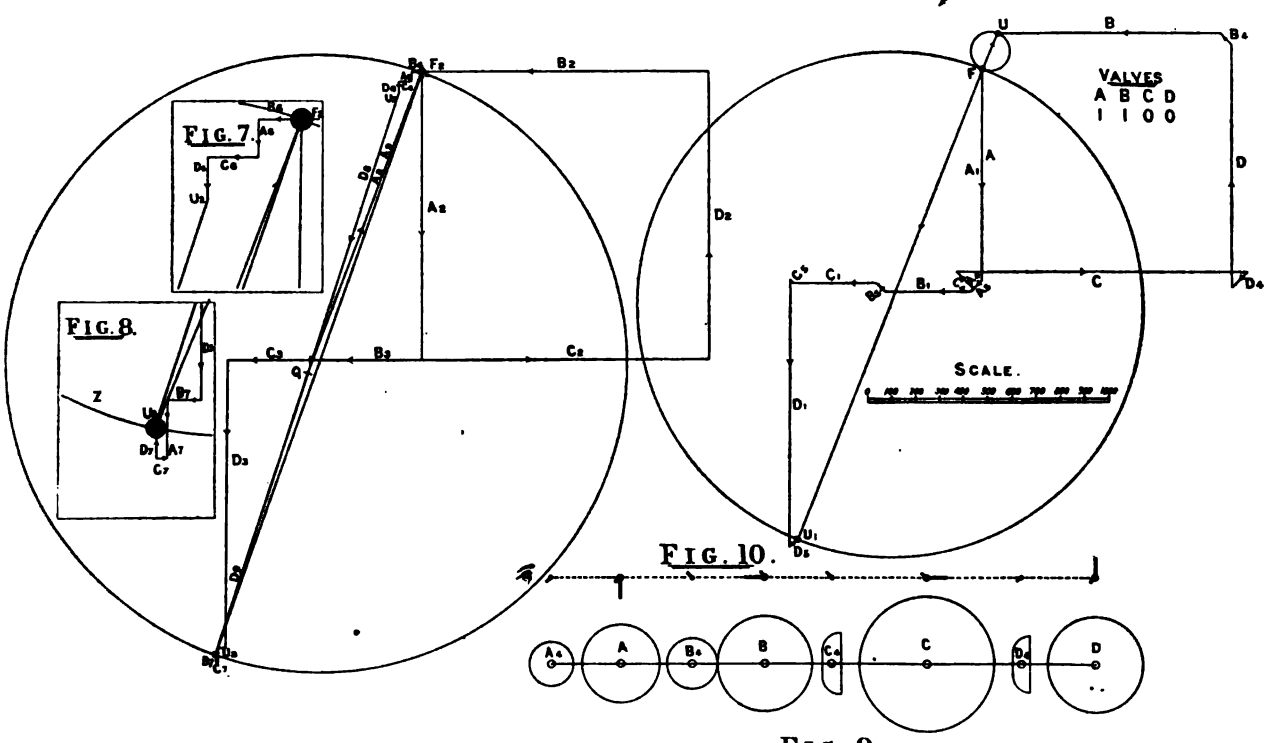


FIG. 9.

The weights that would be necessary to make a balance can be ascertained by drawing, in Fig. 6, the line U 2 U 3, bisect in Q, and draw Q F. Mark U2 Q as D 8, and Q F as A 8 and A 9. The gap U2 F is now made up by the weights U2 Q and Q F; and the moments gap U3 F is made up by U3 Q, the moment of D 8, and Q F, which is A 9, the moment of A 8. Observe that D 9 is the same length as D, because D is at unit distance, and it is oppositely directed, because D is on the aft side, or negative side, of the central plane. The moment A 9 is coincident with the weight A 8,

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and in the same direction, because A is on the forward, or positive, side of the negative plane. The directions are throughout indicated by darts on the lines, and the lettering is intended to be expressed in the vector sense, U3 Q being the opposite of Q U3. The A 8 and D 8 here represented are, of course, quite impracticable as balance weights; A 8 would in this instance be 18 tons in weight, and D 8 would be 16½ tons. The obliquity effect is, however, in this example practically *nil*.

No. 2.—Obliquity Effect.

| | A | B | C | D | Sum. | ÷ 4 | Hyp. | Virtually. |
|---|---|---|---|---|---|---|---|---|
| Cos 2 | 876 | − 910 | − 1,205 | 1,006 | − 233 | 58 | 58 | 14 |
| Sin 2 | 0 | 0 | 0 | 0 | 0 | 0 | | |
| | A 1 | B 1 | C 1 | D 1 | | | | |
| Cos¹ 2 | 876 | − 354 | 349 | − 1,006 | − 135 | 34 | 34 | 8 |
| Sin¹ 2 | 0 | 0 | 0 | 0 | 0 | 0 | | |

To convert unified expressions for resultant forces, or resultant moments, to common measure, say, tons and foot-tons, or kilogrammes and kilogrammetres, or any other standard, multiply by—

$$\frac{N^2 R W}{91{,}200\, g} \text{ for forces,}$$

or by—

$$\frac{N^2 R W L}{91{,}200\, g} \text{ for moments.}$$

Here—

N = revolutions per minute.
R = length of crank.
W = mean piston-line-reciprocating weight.
L = half the distance between the centres of the extreme cylinders.
$g$ = gravitation accelerity.

These may be in any units. When lengths are in feet and $g$ is 32·2, the denominator is 2,937,000.

In the example No. 2 this reduces to—

Divide by 42 for forces in tons.
Divide by 2·7 for moments in foot-tons.

The combined resultant lifting force 65 amounts, therefore, to only 1·5 tons, and the combined vertical longitudinal tilting 4,660, divided by 2·7 gives 1,726 foot-tons.

The latter is equivalent to a pair of equal forces, each 55 tons, acting vertically in opposite directions, harmonically, at the extreme cylinders, stroke for stroke with the

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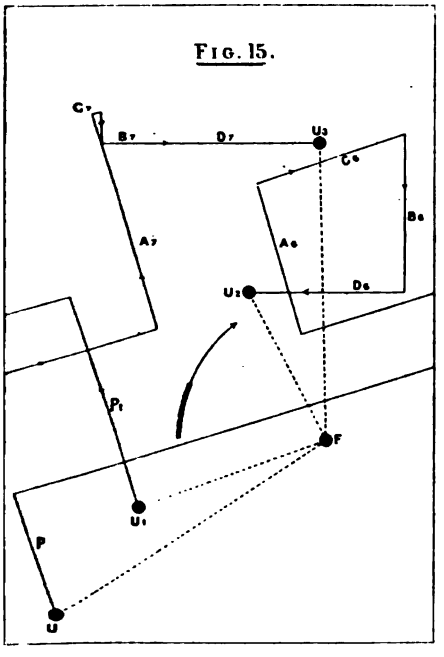
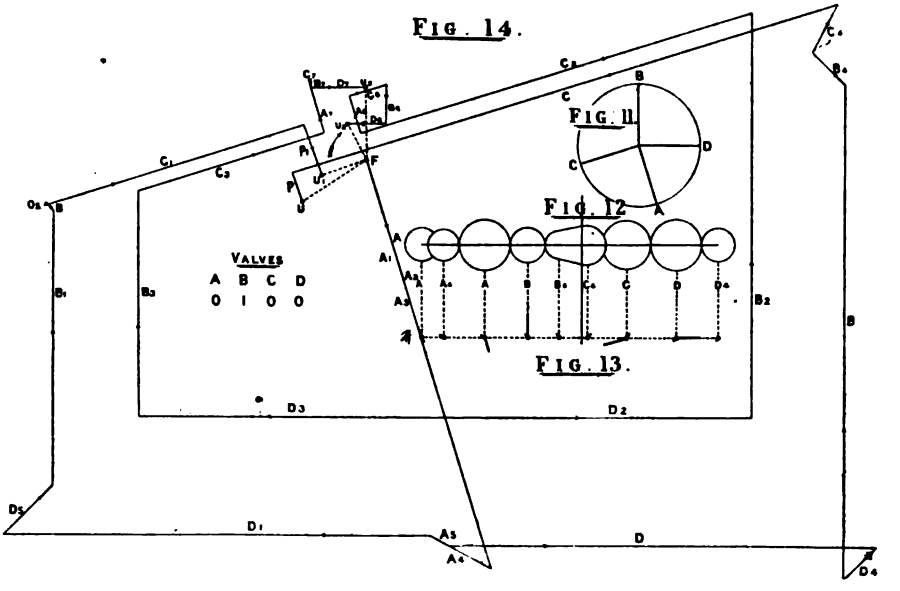
engine. These features were known to the makers when they supplied the data to me. They have since sent me particulars for a more modern engine which is completely balanced. I do not give it in this paper.

No. 3.—Unified Data.—Figs. 11 to 15.

| | Cylinder Arms. | 0 | 1 | 2 | 3 | Valve Arms. | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|---|---|
| A | 1· | 890 | 890 | 560 | 560 | 1·432 | 100 | 143 | 74 | 106 |
| B | ·555 | 1,032 | 573 | 845 | 470 | ·23 | 94 | 22 | 74 | 17 |
| C | ·477 | 1,190 | 567 | 855 | 407 | ·06 | 114 | 7 | 74 | 5 |
| D | 1· | 890 | 890 | 640 | 640 | 1·432 | 100 | 143 | 74 | 106 |

This is an engine approximately on the Schlick plan, but the particulars as supplied to me do not combine exactly. It is instructive to observe how slight variations of weights and angles affect the resultants, therefore, the diagram has been




made first with the data received, and another diagram will be shown with the data probably as designed. There is an air pump at the fore end, its unified reciprocating weight 66 and moment 110. The valves are outside, inside, outside, outside.

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The maximum resultants, in unified measure, are as follows:—

F U = 160, reciprocating weight.
F U2 = 80, revolving weight.
Combined = 170, vertical force.
F U1 = 100, reciprocating moment.
F U3 = 150, revolving moment.
Combined = 150, vertical moment.
F U2 = 80, horizontal force.
F U3 = 150, horizontal moment.

The combined resultants are not drawn, they are obtained by completing the parallelogram on the components, say, on F U and F U2, and measuring the diagonal from F = 170.

No. 4.—Figs. 16 and 17. Modified Data.

| | Cyl. Arms. | 0 | 1 |
|---|---|---|---|
| A ... ... | 1· | 870 | 870 |
| B ... ... | ·552 | 1,032 | 570 |
| C ... ... | ·478 | 1,150 | 550 |
| D ... ... | 1· | 890 | 890 |

The other particulars are the same as in No. 3. The initial point F is moved to the left from the former position marked by a cross, A is diminished by 20 and C by 40. The centres of cylinders B and C are shifted—in ·003 and ·001 respectively. The resultants are now *nil* for the reciprocating weights. The revolving weights produce a vertical force 115, and an equal horizontal force; and a vertical moment 165, with an equal horizontal moment. The vertical force has been reduced 55, and the horizontal force increased 35. The gross result is—

| | No. 3. | No. 4. |
|---|---|---|
| F U2 = Vertical force ... ... ... ... | 170 | 115 |
| F U3 = Vertical moment ... .. ... ... | 150 | 165 |
| F U2 = Horizontal force ... ... ... ... | 80 | 115 |
| F U3 = Horizontal moment ... ... ... ... | 150 | 165 |

This comparison shows that the total results are about equivalent. No. 4, however, admits of an easier application of balance weights to complete the primary balancing, were that considered to be requisite. The weights might be either as in Fig. 17 or as in Fig. 17A. In Fig. 17 there is U2 Y = 10 at D, producing a moment U3 Z, these are D 8 and D 9. The position is D, because the direction of U2 Z is opposite to the direction of U3 Z. The darts are not shown, but the direction is indicated by the

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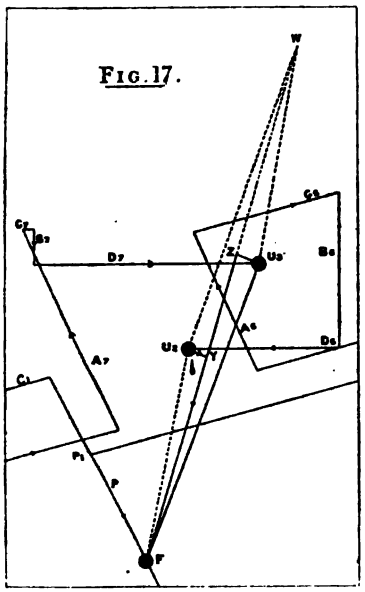
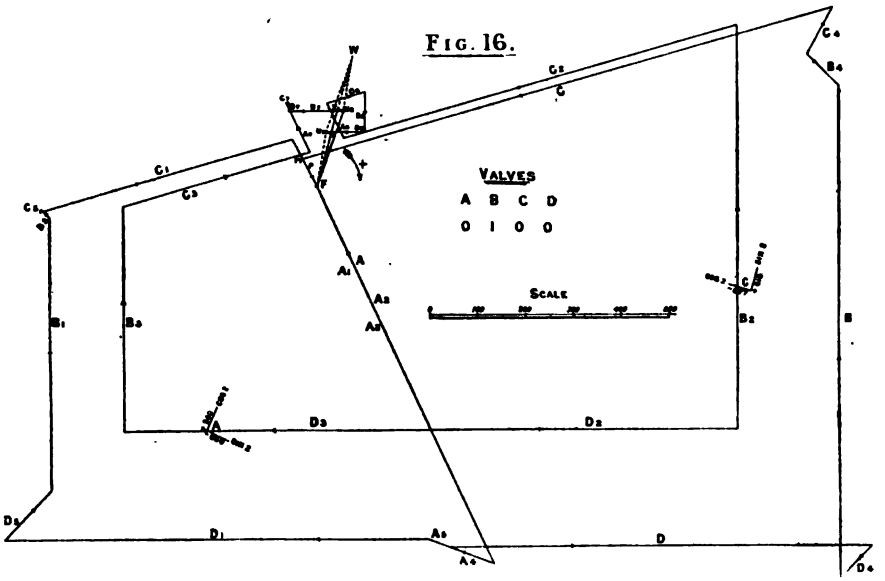
sequence of the other lines. There are also two more weights, one forward and the other aft, the forward one the heavier; the difference = Y F = 115, placed so that their combined moment is U3 F = 165. The polygons are closed by these, U3 F by U3 Z and Z F, and the gap U2 F by U2 Y and Y F.

A more direct way of closing the gaps is that shown in Fig. 17A. Join U2 U3, bisect in Q, and join Q F. The line U2 Q represents D 8, a weight = 31 at D, and U3 Q is D 9, the moment of D 8. The line Q F is A 8 and A 9, a weight = 138 at A, and its moment the same in length and in direction. Observe that the weight U2 Q is at D, because U3 Q has opposite direction and is of the same length. Also Q F is a




weight at A, because its moment, also Q F, has the same direction as the line representing its weight. In 17A the weight U2 Q is not restricted to the position D. The point Q may be anywhere in U2 U3, but the distance of this weight from the central plane must always be Q U3 ÷ U2 Q.

The obliquity effect for inclined piston lines as A and C (Fig. 16) is conveniently found thus:—Draw a line at the reverse angle from either end of the piston line, using a jointed sector and a tee-square; then, shifting the square to the other edge of the drawing-board, apply the sector again to complete the right-angled triangle upon the piston line as hypotenuse. Mark only the right angle corner, as at C. The length along the reversed-angle line is C cos 2 $c$, the other side is C sin 2 $c$. A 45° set-square applied to the piston line shows in which octant it lies, and the signs are as marked on the semicircle here drawn.

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If lines, as in Fig. 17B, are drawn upon a 45° celluloid set-square, it will be found very convenient in working the problem of obliquity effect. The set-square must be turned the right side up for the direction of rotation. In the first octant on the left the sine and the cosine are both negative. In the second octant the sine is alone

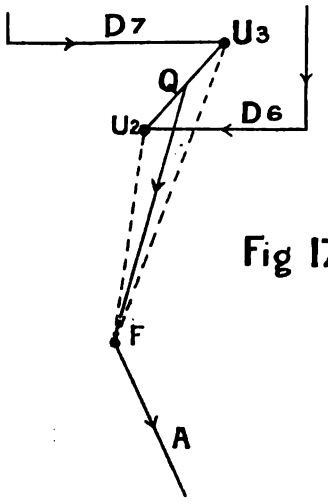


Fig 17A

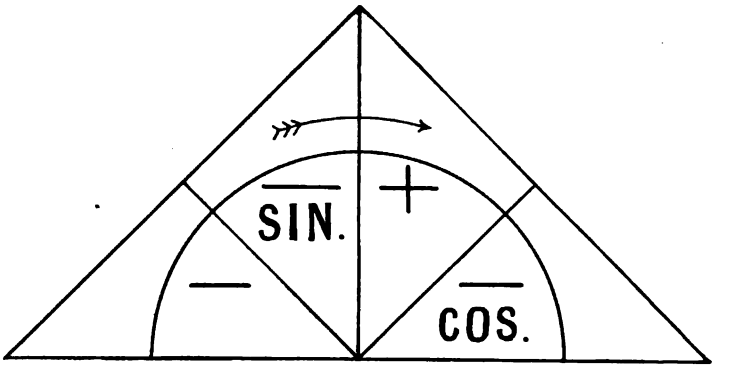


FIG. 17B.

negative. In the third octant both are positive. In the fourth octant the cosine alone is negative. There are other guides to the proper signs, but this is the safest and simplest known to me. The signs are read irrespectively of the directions of the dart upon the line. The centre of this semicircle is placed upon the line with the hypotenuse of the set-square horizontal. Tabulating these from Fig. 17, we get—

NO. 4.—OBLIQUITY EFFECT.

| | A | B | C | D | Sum. | ÷ 4 | Hyp. | Virtually. |
|---|---|---|---|---|---|---|---|---|
| Cos 2 | 560 | 1,032 | – 977 | – 890 | – 275 | – 69 | 70 | 17 |
| Sin 2 | – 660 | 0 | 618 | 0 | – 42 | – 10 | | |
| | A 1 | B 1 | C 1 | D 1 | | | | |
| Cos¹ 2 | 560 | 570 | 466 | 890 | 2,486 | 621 | 675 | 169 |
| Sin¹ 2 | – 660 | 0 | – 295 | 0 | – 955 | – 239 | | |

The obliquity resultant for vertical force is very small, only 70; but the obliquity tilting is great, 675.

The effect of the slide valve gear in the balancing result has not been specially dealt with in any of the papers we have had on balancing. Professor Dunkerley made

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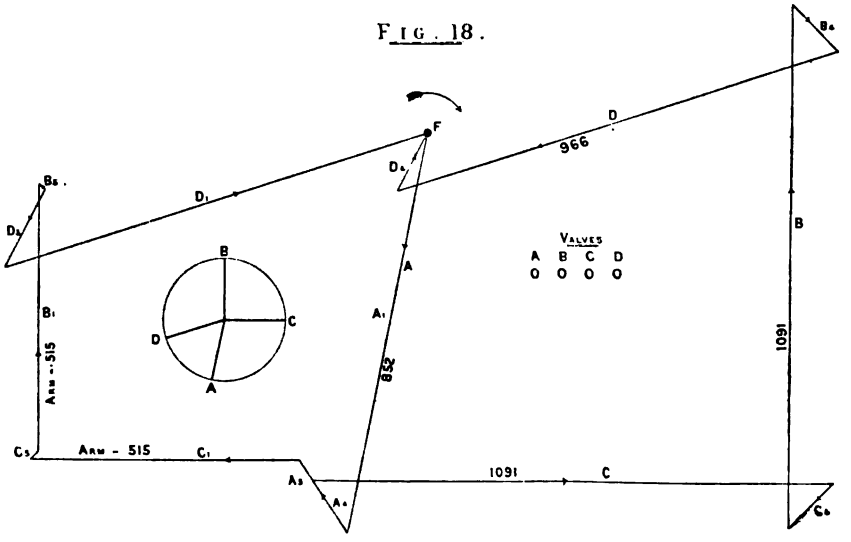
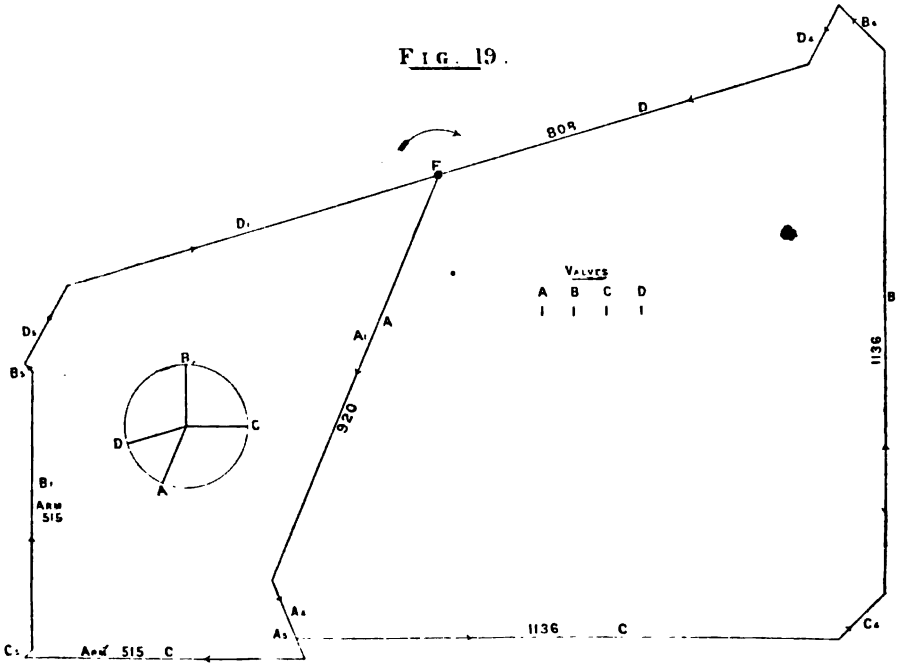
some remarks on this in the discussion on Herr Otto Schlick's paper last year (page 158). I have, therefore, made some diagrams to show the effect of the valve gear, especially the influence of the slide valves as they are outside valves or inside valves.

FIG. 18.




In the three examples next given, the centres and the valve gear weights are as in No. 1. The reciprocating weights alone are represented, as these are the important

FIG. 19.




weights, not easily admitting of ameliorating balancing by the addition of revolving weights. The object of the diagrams is to determine what are the proportions of the



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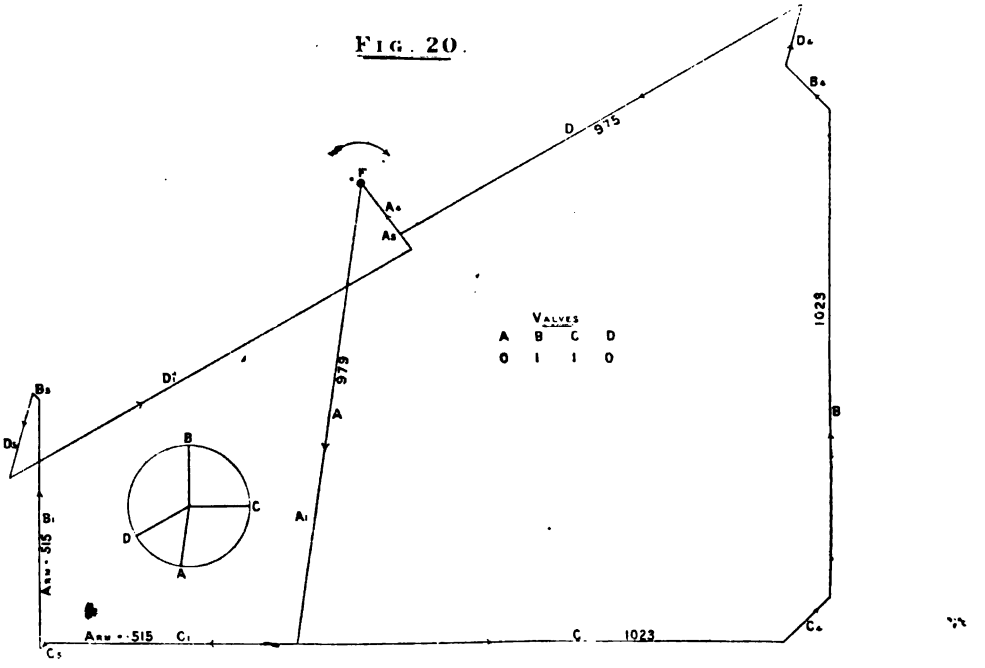
piston-line-reciprocating weights that will give closed polygons, and the centres and inclinations of the cranks.

In balancing a four-crank engine according to the Schlick system, it is a characteristic feature of the construction that the moving elements of the two inner cylinders must be heavier than those of the outer cylinders. This difference is so great that it is impracticable on this system to have the forgings and brasses for the four cylinders duplicates. The rule he gives for dealing with the valve gear is to make the distances between the valve rods, the weight of the moving parts of the valve gear, the radii and angles of the eccentrics to have the same relation to one another as the

FIG. 20.




moving elements of the cylinders and cranks. This ensures that the same disparity of weights will be maintained throughout the combined system of valve gear and piston-line weights, which is the characteristic of the system of piston-line weights by themselves.

This mode of dealing with the valve gear precludes altogether the application of any modifying influence latent in the valve gear. In some classes of steamships it is desirable to limit this disparity of weights to such differences as can conveniently be made by adding to the weights of piston castings, so that the spare gear carried will be identical for all the cylinders. To exhibit the influence of the valve gear weights, the examples 5, 6, 7 are given, with the same centres and the same valve gear weights as in No. 1, which represents an actual engine of the class referred to. The piston-

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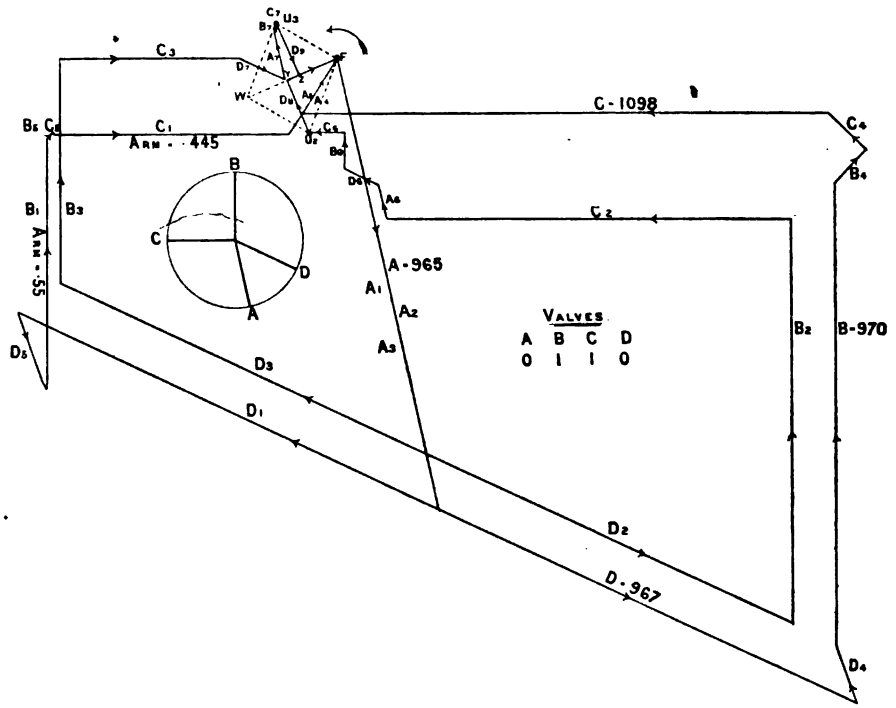
line weights are here tabulated, along with those of No. 4. The type of valve for each cylinder is given in the heading of each column.

| | No. 4. A B C D O I O O | No. 5. A B C D O O O O | No. 6. A B C D I I I I | No. 7. A B C D O I I O |
|---|---|---|---|---|
| A ... | 870 | 852 | 920 | 979 |
| B ... | 1,032 | 1,091 | 1,136 | 1,023 |
| C ... | 1,150 | 1,091 | 1,136 | 1,023 |
| D ... | 890 | 966 | 808 | 975 |

These are given in unified measure: the mean piston-line-reciprocating weight is 1,000 in each column. It is remarkable how the disparity is reduced when the inner

FIG. 21.




cylinders have inside valves and the outer cylinders have outside valves. In No. 7 the two outer cylinders have the piston-line weights only 44 and 48 lbs. less than those of the inner cylinders, or less than 5 per cent. of difference.

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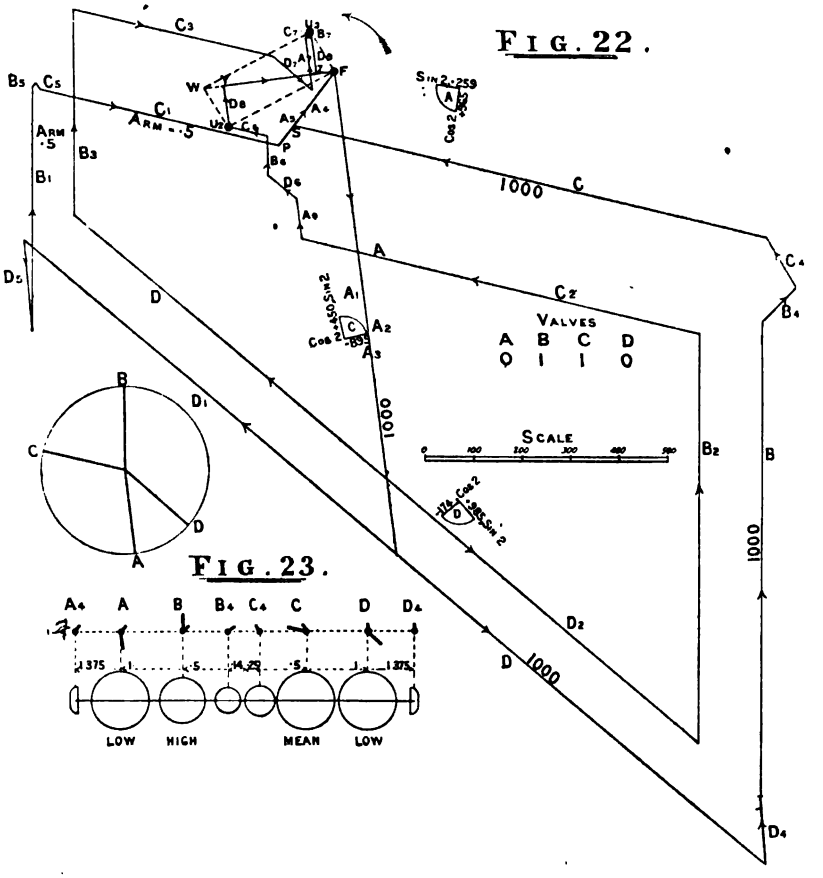
Examples No. 8 and No. 9, shown in Fig. 21 and Fig. 22, carry this principle still further. As the centres and weights are somewhat different from those in the preceding examples, the complete tables of elements are given. These are drawn as for rotation right to left.

No. 8.—Unified Data.—Fig. 21.

| | Cylinder Arms. | 0 | 1 | 2 | 3 | Valve Arms. | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|---|---|
| A | 1· | 965 | 965 | 840 | 840 | 1·4 | 130 | 182 | 75 | 105 |
| B | ·55 | 970 | 531 | 840 | 462 | ·22 | 90 | 19·8 | 75 | 16 |
| C | ·445 | 1,098 | 487 | 840 | 374 | ·044 | 110 | 4·4 | 75 | 3·3 |
| D | 1· | 967 | 967 | 840 | 840 | 1·4 | 130 | 182 | 75 | 105 |




The valves are O I I O in No. 8. The high-pressure cylinder would be B, the intermediate C, and the two low pressures would be A and D.

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In No. 9, by reducing the angle between the inner cranks, the inequality of weights has entirely disappeared, the valves being outside, inside, inside, outside.

No. 9.—Unified Data.—Figs. 22 and 23.

| | Cylinder Arms. | 0 | 1 | 2 | 3 | Valve Arms. | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|---|---|
| A | 1· | 1,000 | 1,000 | 840 | 840 | 1·375 | 140 | 192 | 80 | 110 |
| B | ·5 | 1,000 | 500 | 840 | 420 | ·14 | 100 | 14 | 80 | 11 |
| C | ·5 | 1,000 | 500 | 840 | 420 | ·125 | 120 | 15 | 80 | 12 |
| D | 1· | 1,000 | 1,000 | 840 | 840 | 1·375 | 140 | 192 | 80 | 110 |

No. 9.—Obliquity Effect.

| | A | B | C | D | Sum. | ÷ 4 | Hyp. | Virtually. |
|---|---|---|---|---|---|---|---|---|
| Cos 2 | 965 | 1,000 | − 895 | − 174 | 896 | 224 | 479 | 120 |
| Sin 2 | 259 | 0 | 450 | 985 | 1,694 | 423 | | |
| | A 1 | B 1 | C 1 | D 1 | | | | |
| Cos¹ 2 | 965 | 500 | 447 | 174 | 2,086 | 521 | 573 | 143 |
| Sin¹ 2 | 259 | 0 | − 225 | − 985 | − 951 | 238 | | |

The effect of obliquity has, however, greatly increased in the vertical force. If Herr Otto Schlick's views are correct on the effect of secondary forces and secondary couples, the neutralisation of these obliquity effects "is not of very great importance."

I intended to have gone over more ground when I gave the title to this paper, but I became otherwise engaged.

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## DISCUSSION.

Mr. C. H. Wingfield (Member): Gentlemen, I am afraid it is difficult to discuss a paper of such complexity as this without very careful study, which, of course, I have not had an opportunity of giving to it. I may say, however, that I have seen something of the author's methods, and have found them very practical. Mr. Macfarlane Gray has made one very great departure, which he has mentioned at the end of the paper; indeed, the postscript is really, I think, one of the most important parts of his paper. It is very desirable, especially in Government work, to minimise the number of spare engine parts which have to be carried on a ship. In engines for H.M. Navy

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there has to be a spare piston rod of every size, a spare connecting rod of every size, and spare eccentric rods of every size; and, if you can make them all the same size, you may get one-third or one-fourth the quantity of spare gear, and you considerably decrease the weight to be carried on board as well as the cost of providing it. By Mr. Macfarlane Gray's method of making use of the valves themselves and the valve gear in order to get the balance, he has gone a very long way indeed towards helping us in that way. The only other method of balancing by adjusting the crank angles in general use is that introduced by Herr Schlick. In order to apply the Schlick method of balancing, however, "the moving elements of the two intermediate cylinders *must* be heavier than those of the outer cylinders (and *this is a characteristic feature* of this construction"). The words I have italicised in the above quotation from Herr Schlick's patent specification show clearly what a great advance Mr. Macfarlane Gray has made. He has succeeded in putting the same weights on the outer cranks as on the inner ones. They would be a great deal too heavy according to Herr Schlick's method; but when the piston jumps up Mr. Gray makes the valve jump down, and in that way the effect of the jump up is partly neutralised and the resultant effect is no more than if he had got a very light piston. Another point shown here for the first time is the exceedingly neat method of allowing for the effect of the valve gear and the crank-weights simultaneously. Mr. Macfarlane Gray last year described a simple graphical method in which he took only the piston rods and the connecting rods into consideration, and left out the valve gear. He says that the simultaneous consideration of the positions of cranks and eccentrics is really a drawing office method, but it is a most important thing in applying his invention. I offer Mr. Macfarlane Gray my thanks for bringing this important paper before us.

Mr. J. Macfarlane Gray (Member of Council): My Lord, I am pleased to hear Mr. Wingfield's remarks on my paper. He has a previous knowledge of it. When others try to carry out the methods described by me in the drawing office, they will also find them easy to understand and simple in application. The lettering on the drawings had, unfortunately, been made too small to allow for reduction by photography, and the members had, therefore, been unable to follow the paper whilst it was read, or more criticism might have been expressed.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): The gentleman who has just sat down congratulated Mr. Macfarlane Gray on his paper, and I believe you will all agree in thanking him for the very interesting paper he has just read.

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# ON THE BALANCING OF THE RECIPROCATING PARTS OF ENGINES, INCLUDING THE EFFECT OF THE CONNECTING ROD.

By Professor W. E. Dalby, M.A., B.Sc., Associate.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 28, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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At the Spring Meeting of the Institution in 1899, I had the honour to communicate a paper, giving a direct and simple method of balancing engines, assuming that the reciprocating parts moved with simple harmonic motion. The method depended partly upon computation, partly upon drawing, the computation being facilitated by the use of certain schedules, arranged so that the work could be done in an orderly fashion. It will be remembered that, considering a revolving system of masses, the fundamental idea of the method was the fixing of *one* point on the axis of revolution, and the consideration of the actions of all the centrifugal forces with respect to it, each force giving rise to an equal, and parallel force, acting on the point, and to a couple. The forces at the point all lie in a plane, at right angles to the axis of revolution, which plane was materialised into a sheet, or drawing board, keyed to, and revolving with, the shaft, and called a Reference Plane; and it was in this plane that the forces and the axes of the couples were supposed to be combined into closed polygons. The properties of these closed polygons conditioned the balancing. It will also be remembered that the same method was extended to a system of masses reciprocating with simple harmonic motion, and that the problem of balancing an engine really consisted of two parts: the first being concerned with balancing the reciprocating masses amongst themselves, fixing thereby the crank angles; the second, with the determination of the proper balance weights for the crank shaft, to suit the crank angles and cylinder pitches, fixed by the solution of the first part of the problem. Certain rules were also given with respect to the selection of the data in any proposed problem.

The present paper is wholly concerned with balancing reciprocating masses amongst themselves, taking into account the variation of the simple harmonic motion caused by the connecting rods. Briefly, it is shown, in Articles 5 to 7, that the

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balancing is conditioned now by four polygons, two force and two couple polygons, this being established by using the approximate expression (2), Article 2, for the acceleration of the crosshead masses. This expression has been used by Mr. Mallock, by Mr. Mark Robinson, and Captain Sankey; it is the basis of Mr. Macfarlane Gray's Accelerity Diagram; it has been used by M. Normand, and, more recently, by Herr Schlick, in his interesting paper of last spring. The degree to which it approximates to the real acceleration is examined in Article 3. The error is small, and its effect is, probably, very much less than the effects of the unbalanced auxiliary engines, or of a propeller, even slightly out of balance. The essential feature of this paper is the merging of the simple geometrical ideas of my first paper into an analytical method, by means of which the possibilities of balancing any particular engine, or the estimation of the unbalanced forces and couples, due to a given arrangement, may be investigated, including the effect of the connecting rod, without the use of trigonometry. In fact, the tentative methods of trigonometry are replaced by the more direct methods of algebra. This all comes of the way used to specify the direction of the sides of the conditioning polygons.

Instead of defining the direction of a line by the angle $\theta$, which it makes with an initial line, the rectangular co-ordinates of two points are used to specify the direction, one point being at the origin, in the way explained in Article 8. The lines forming the sides of the polygons are, in fact, treated as vectors by an analytical process. The result of applying this method to fix direction is to considerably simplify the analytical treatment of the balancing problem in every particular case. Incidentally, I have elaborated the question of data, and have shown what conditions operate to limit the free selection of the fixable quantities, in Articles 12 to 14. After establishing the general method, it has been applied to several actual examples, including the case of partial balancing, given in Herr Schlick's paper (Trans. I.N.A., 1900, Vol. XLII.). The solution for the balancing of a four-crank engine completely is given in Article 19, and, though it cannot be applied practically, it is used, in Articles 23 and 24, to obtain new solutions, with respect to the balancing of five and six crank engines, which solutions are shown to satisfy four more conditions in Article 25. The latter part of the paper is devoted to the estimation and computation of errors in certain typical cases.

Dr. W. E. Sumpner, Principal of the Municipal Technical School, Birmingham, has taken much trouble in revising the proofs, and, while thanking him, I should like to acknowledge the help I have gained from his criticisms and suggestions.

## 2. Analytical Expression for the Acceleration of the Piston, including the Effect of the Connecting Rod.

Let $\theta$ be the variable angle between a fixed line of reference O Z (Fig. 1, Plate XLII.), the centre line of the engine, say, and a line of reference O $X_1$, drawn in the

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revolving reference plane, containing the crank O P. The circle may be looked upon, in fact, as a crank disc, on which the lines O P and O $X_1$ are scribed.

Let $a$ be the constant angle between the direction of the crank O P, and the line of reference O $X_1$; $\phi$, the angle between the connecting rod and O Z; $r$ the crank radius; $l$ the length of the connecting rod.

The distance, $x$, of the crosshead, B, from the origin O, is given by the expression—

$$r \cos(\theta + a) + l \cos \phi.$$

But, since—

$$r \sin(\theta + a) = l \sin \phi,$$

$$\cos \phi = \sqrt{1 - \frac{r^2}{l^2} \sin^2(\theta + a)}, \qquad (a)$$

$$\therefore \quad \cos \phi = 1 - \frac{r^2}{2 l^2} \sin^2(\theta + a), \text{ approximately;}$$

$$\therefore \quad \cos \phi = 1 + \frac{r^2}{4 l^2} \{\cos 2(\theta + a) - 1\}.$$

Therefore—

$$x = r \cos(\theta + a) + \frac{r^2}{4 l} \cos 2(\theta + a) + \left(l - \frac{r^2}{4 l}\right). \qquad (1)$$

Differentiating twice, with respect to the time, and considering the angular velocity, $\dot{\theta} = \omega$, of the crank to be sensibly constant, the acceleration of B is—

$$\frac{d^2 x}{d t^2} = - r \dot{\theta}^2 \cos(\theta + a) - \frac{r^2 \dot{\theta}^2}{l} \cos 2(\theta + a).$$

Let M be the mass reciprocated by the point B, then the instantaneous value of the unbalanced force, acting on the engine frame in the line of stroke, which is equal and opposite to the force required for M's acceleration is given by—

$$M \omega^2 r \left\{ \cos(\theta + a) + \frac{r}{l} \cos 2(\theta + a) \right\}. \qquad (2)$$

### 3. On the Error Involved by the Approximation.

It will be observed that the approximate formulæ (1) and (2), of Article 2, depend upon the extraction of the square root of the expression giving $\cos \phi$, by the Binomial Theorem, to two terms only, it being tacitly concluded that the remaining terms in the expansion may be neglected, without involving serious error. How surprisingly small the error is may be seen by comparing the real acceleration for a few crank positions, with the acceleration calculated by the approximate formula (2). Differentiating the true expression for the displacement $x$, twice, with respect to the time, the true value of the acceleration is given by—

$$\omega^2 r \left\{ \cos(\theta + a) + \frac{r l^2 \cos 2(\theta + a) + r^3 \sin^4(\theta + a)}{\{l^2 - r^2 \sin^2(\theta + a)\}^{\frac{3}{2}}} \right\} \qquad (3)$$



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The values calculated by this formula are compared with the approximate values calculated by formula (2), for 30° intervals, in the following Table, for a rod $3\frac{1}{2}$ times the length of the crank. It will be noticed that when $(\theta + \alpha) = 0°$, or 180°, the values given by the formula (2) are exact, because $\sin(\theta + \alpha) = 0$, and, therefore, the true expression reduces to the same form as the approximate one.

The greatest percentage error in the Table is at 90°, and is of the order of 1 per cent.

TABLE I.

| Crank Angle. | True Acceleration. | Approximate Acceleration. |
|---|---|---|
| Degrees. | | |
| 0 | + 1·286 | + 1·286 |
| 30 | + 1·0148 | + 1·009 |
| 60 | + ·3571 | + ·357 |
| 90 | − ·2981 | − ·286 |
| 120 | − ·6429 | − ·643 |
| 150 | − ·7172 | − ·723 |
| 180 | − ·714 | − ·714 |
| 210 | − ·7172 | − ·723 |
| 240 | − ·6429 | − ·643 |
| 270 | − ·2981 | − ·286 |
| 300 | + ·3571 | + ·357 |
| 330 | + 1·0148 | + 1·009 |
| 360 | + 1·286 | + 1·286 |

4. Graphical Interpretation of the Expression (2), Article 2.

The first term of this expression—

$$M\,\omega^2\,r\cos(\theta + \alpha),$$

is equal to the projection O A (Fig. 1), on the line of stroke, of the centrifugal force, due to a mass M, concentrated at the crank radius.

Multiply, and divide the second term by 4, giving—

$$M\,(2\,\omega)^2\left(\frac{r^2}{4\,l}\right)\cos 2\,(\theta + \alpha).$$

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This is equal to the projection O B, on the line of stroke, of the centrifugal force due to M, concentrated at a crank radius $\frac{r^2}{4l}$ rotating twice as fast as the main crank.

In this way, the unbalanced force caused by the reciprocation of M may be separated into a primary and a secondary part.

The primary part is simply the projection, on the line of stroke, of the centrifugal force, due to the reciprocating mass, supposed transferred to the crank pin.

The secondary part is the projection, on the line of stroke, of the centrifugal force, due to the reciprocating mass, supposed transferred to the crank pin of an imaginary crank, $\frac{r}{4l}$ times the radius of the main crank, revolving in the same plane twice as fast as the main crank.

The line of reference O $X_1$ (Fig. 1) revolves at the same speed as the crank; the angle $\alpha$, therefore, remains constant. Similarly, the line of reference O $X_2$, revolves twice as fast as the main crank, and the angle $2\alpha$ remains constant.

5. The Effect of the primary and secondary unbalanced Forces with respect to a Reference Plane $a$ feet from the Plane of Revolution of the Crank.

Fig. 2, Plate XLII., shows a crank, and the instantaneous relative position of the imaginary crank causing the secondary forces. The effect of the two cranks, with reference to any chosen reference plane, is conveniently treated by supposing that there are two coincident reference planes: one belonging to the main crank, and revolving as though keyed to the shaft; and one belonging to the secondary crank, and revolving twice as fast as the shaft. The two planes may be thought of as a pair of infinitely narrow fast and loose pulleys. These planes are shown separated in Fig. 2, for the sake of clearness. The principles of the method I explained in my previous paper may now be applied to the consideration of this double effect. Due to the mass M, at the main crank radius $r$, whose angular velocity is $\omega$, there is—

(1) A force $M\omega^2 r$, equal and parallel to the centrifugal force due to M, shown by O $a$, to scale, in No. 1 reference plane;

(2) A couple, whose moment is $M\omega^2 r a$, represented by O A in No. 1 plane.

Due to the mass M, at the imaginary crank radius $\frac{r^2}{4l}$, whose angular velocity is $2\omega$, there is—

(1) A force $M\omega^2 r^2/l$, equal and parallel to the centrifugal force due to M, and shown by $O_1 s$, in plane No. 2;

(2) A couple, whose moment is $M\omega^2 r^2 a/l$, and shown by $O_1$ S, in plane No. 2.

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### 6. Effect of more than one Crank on the same Shaft.

If there are several cranks on the same shaft, each will be accompanied by its imaginary fellow, and each crank will, therefore, give rise to a set of forces, and couples, in a given pair of coincident reference planes, similar to the set stated above. The angles between the set of imaginary cranks will remain constant, although the shaft is revolving twice as fast as the main crank; in fact, they form an imaginary crank shaft, in which the angles between any pair of the imaginary cranks is always double the angle between the corresponding pair of real cranks. If the effect of each crank is referred to one pair of reference planes, as in Fig. 2, the whole effect will be represented by the vector sums of the several forces and couples. The sum of the projections of the resultant force vectors on the line of stroke will give, at any instant, the value of the disturbing force; and the sum of the projections of the resultant couple vectors, the value of the disturbing couple. Evidently the conditions that there shall be no force and couple are that the four polygons shall separately close. For instance, if there were four cranks on the shaft, Fig. 2, and, if it were possible to draw the closed polygons—

$$\mathrm{O\,A\,B\,C},\ \mathrm{O}\,a\,b\,c;\ \mathrm{O_1\,S\,T\,U},\ \mathrm{O_1}\,s\,t\,u,$$

the engine would be balanced both for primary and secondary forces and couples; because, obviously, their several projections on the line of stroke would be zero for all positions of the polygons, that is, for all values of $\theta$.

### 7. The Conditions of Balance.

The conditions of balance, to include the effect of the connecting rod, are, therefore, completely stated thus. Choosing a pair of coincident reference planes anywhere along the shaft—

$$\left.\begin{array}{l}\text{(1) The primary force polygon must close.}\\ \text{(2) The primary couple polygon must close.}\\ \text{(3) The secondary force polygon must close.}\\ \text{(4) The secondary couple polygon must close.}\end{array}\right\}\ \mathbf{B.}$$

### 8. Analytical Representation of a Vector Quantity.

Take a pair of axes O X, O Y (Fig. 3, Plate XLII.). Let $\mu$ represent a vector, whose magnitude is O V. The direction of the vector is determined by the two quantities $x$ and $y$, measured along O X, and parallel to O Y, respectively, having regard to the usual convention respecting signs. The end of $y$, remote from the X axis, fixes a point $q$. The line joining O to $q$ defines the direction of the vector completely.

There are an infinite number of pairs of values of $x$ and $y$, $\frac{y}{x}$ being constant, which will define the same direction, and the choice of a pair depends upon the

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particular work in hand. If the direction is to be set out on a drawing, $x$ and $y$ should be chosen to bring $q$ as far from O as possible, to ensure accuracy. For purposes of analytical investigation, it is usually more convenient to choose that pair of values which make

$$x^2 + y^2 = 1\,;$$

in which case the vector is represented by

$$\mathrm{O\,V}\,(x + i\,y)\,;$$

$i$ may be looked upon as a symbol of operation, directing that $y$ is to be set out parallel to the axis of Y, *from* the axis O X *towards* $q$. Considering the figure, it is evident that, assuming

$$x^2 + y^2 = 1,$$

it follows that $x = \cos a$, and that $y = \sin a$; also, if $x = 0$, then $y = 1$; and, if $y = 0$, $x = 1$, since $x^2 + y^2$ is, by hypothesis, always equal to unity.

The magnitude of O V is always a positive quantity. Further, if, in the two vectors, O V $(x + i\,y)$ and $\mathrm{O_1\,V_1}\,(x_1 + i\,y_1)$, $x = x_1$, then $\pm\,y = \pm\,y_1$; the sign of the $y$ in each case being determined by the conditions of the problem.

### 9. Relation between the Quantities defining the Directions $a$ and $2\,a$.

Let $x$ and $y$ (Fig. 4, Plate XLII.) define the direction O $q$, and $x_1$, $y_1$, the direction O $q_1$. The four quantities are connected by the condition that the angle X O $q_1$ is to be double the angle X O $q$. Since

$$\mathrm{X\,O}\,q_1 = 2\,a,$$

and—

$$x_1 = \cos 2\,a = \cos^2 a - \sin^2 a,$$

$$x_1 = x^2 - y^2\,;$$

since $x$ and $y$ are respectively the cos and sin of $a$; similarly—

$$y_1 = \sin 2\,a = 2 \sin a \cos a\,;$$

$$\therefore\quad y_1 = 2\,x\,y.$$

Therefore, if $x$ and $y$ define a direction $a$, $(x^2 - y^2)$, and $2\,x\,y$ are the pair of values of the quantities defining the direction $2\,a$. These are, of course, to be measured along and parallel to O X and O Y, respectively.

### 10. Application to the Balancing Problem.

The magnitude O V, of a force vector of the kind considered in Article 5, is of the form $\mathrm{M}\,\omega^2\,r$, and of a couple vector $\mathrm{M}\,\omega^2\,r\,a$. Hence the side of a primary force polygon is represented by a vector of the form—

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$$M\omega^2 r(x + iy);$$

the side of a primary couple polygon by—

$$M\omega^2 r a(x + iy);$$

the side of a secondary force polygon by—

$$\frac{M\omega^2 r^2}{l}\{(x^2 - y^2) + i\,2xy\};$$

and the side of a secondary couple polygon by—

$$\frac{M\omega^2 r^2 a}{l}\{(x^2 - y^2) + i\,2xy\}.$$

Adding subscripts to distinguish the different cranks, the statement of the conditions B, Article 7, may now be transformed into—

$$\left.\begin{array}{l}
\omega^2 r\{M_1(x_1 + iy_1) + M_2(x_2 + iy_2) + M_3(x_3 + iy_3) \;\ldots\; \} = 0 \\
\omega^2 r\{M_1 a_1(x_1 + iy_1) + M_2 a_2(x_2 + iy_2) + \;\ldots\ldots\; \} = 0 \\
\frac{\omega^2 r^2}{l}\{M_1(x^2_1 - y^2_1 + i\,2x_1y_1) + \;\ldots\ldots\ldots\; \} = 0 \\
\frac{\omega^2 r^2}{l}\{M_1 a_1(x^2_1 - y^2_1 + i\,2x_1y_1) + \;\ldots\ldots\ldots\; \} = 0
\end{array}\right\} \text{C}$$

It is a property of the quantities of the kind considered that, in any one equation of the above type, all the quantities associated with the symbol $i$ must of themselves form an expression equal to nothing; the remaining quantities forming a second expression, also equal to nothing.

## 11. The Eight Fundamental Equations.

Using the principle of the previous article, the equations of Group C become, $\omega$, $r$ and $l$ cancelling out—

$$\left.\begin{array}{lll}
\text{Primary forces vanish} & \{M_1 x_1 + M_2 x_2 + \} \;\ldots\; = 0 & (1) \\
 & \{M_1 y_1 + M_2 y_2 + \} \;\ldots\; = 0 & (2) \\
\text{Primary couples vanish} & \{M_1 x_1 a_1 + M_2 x_2 a_2 + \} \;\ldots\; = 0 & (3) \\
 & \{M_1 y_1 a_1 + M_2 y_2 a_2 + \} \;\ldots\; = 0 & (4) \\
\text{Secondary forces vanish} & \{M_1(x^2_1 - y^2_1) + M_2(x^2_2 - y^2_2) + \} \;\ldots\; = 0 & (5) \\
 & \{M_1(x_1 y_1) + M_2(x_2 y_2) + \} \;\ldots\; = 0 & (6) \\
\text{Secondary couples vanish} & \{M_1(x^2_1 - y^2_1)a_1 + M_2(x^2_2 - y^2_2)a_2 + \} \;\ldots\; = 0 & (7) \\
 & \{M_1(x_1 y_1)a_1 + M_2(x_2 y_2)a_2 + \} \;\ldots\; = 0 & (8)
\end{array}\right\} \text{D}$$

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It should be carefully remembered that every $x$ is connected to the $y$, with the same subscript, by the relation $(x^2 + y^2) = 1$; so that, when any $x$ is known, the corresponding $y$ is known also. A direction is, in fact, completely specified by the value of an $x$, or, of course, the value of a $y$. Although the eight equations appear to involve the four sets of unknown quantities in M, $a$, $x$, and $y$, respectively, they are really in the three sets M, $a$, and $x$, since the $y$'s can everywhere be replaced by a function of the corresponding $x$.

The above eight equations completely state the analytical conditions of balance amongst the reciprocating parts, for an engine with any number of cranks, the cylinders being arranged in the way usual in marine work: *i.e.*, all on one side of the crank shaft, their centre lines being all in the vertical plane, which contains the axis of the crank shaft. The connecting rods are supposed to be equal in length, and the masses are the equivalent masses, reduced to the crank radius.

## 12. On the Relation between the Number of conditional Equations and the Number of Variables.

In the application of these equations to any particular example, the possibilities of a solution are indicated by the following propositions, quoted from "Chrystal's Algebra," pp. 286 and 288, Part I.:—

(1) The solution of a system of equations is in general determinate when the number of equations is equal to the number of variables.

(2) "If the number of equations be less than the number of variables, the solution is in general indeterminate" (that is, several solutions are possible).

(3) "If the number of independent equations be greater than the number of variables, there is in general no solution, and the system of equations is said to be inconsistent."

## 13. On the Number of Variables in Balancing Problems.

(1) As to the magnitudes of the masses.

Equations D, Article 11, may be divided through by any one of the M's without altering the conditions expressed, thereby diminishing the actual number of quantities representing magnitude by unity; or the magnitudes of the M's may be diminished, or increased, in the same proportion all through, without changing the conditions stated; or thinking of the vector polygons, the conditions of balance are independent of the scales to which they are drawn. It follows that the number of independent variables, as

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regards magnitude, is the number of ratios between the several magnitudes, and any one of them. Therefore, if there are $n$ cranks, the number of variables of magnitude is

$$n-1.$$

(2) As to the direction of the cranks.

The most general way of measuring the crank direction is from an arbitrarily fixed line of reference, as $O X_1$, Fig. 1, in the reference plane. This line, however, can always be chosen to coincide with the direction of one of the cranks, so that the number of variables of direction, corresponding to $n$ cranks is

$$n-1.$$

These, however, are not independent, for equations 1, 2, 5, 6, and 3, 4, 7, 8, of set D, form two precisely similar sets, the one in M, the other in M $a$. Therefore, if from the first group $\frac{M_4}{M_1} = A$, A being a function of the $x$'s and $y$'s, $\frac{M_4, a_4}{M_1, a_1}$ will equal A also. Therefore A must be such a number that when multiplied by $\frac{a_1}{a_4}$ it repeats numerically. Obviously, therefore, the directions may not be chosen at will.

(3) As to the pitch of the cylinders.

The number of distances, $a_1$, $a_2$, &c., of the planes of revolution from the reference plane corresponds, in general, with the number of cranks. Equations D, Article 11, may be divided through by any one of the $a$'s, without altering the conditions of balance, showing, as for masses, that the number of independent variables in $a$ is

$$n-1.$$

The whole number of variables, corresponding to $n$ cranks, is therefore

$$3\,(n - 1) \qquad (1)$$

If the cylinder centre lines are fixed, the number of variables is

$$2\,(n - 1) \qquad (2)$$

It should be carefully noticed that the variables in M and $a$ are ratios. The value of any one magnitude may always be put equal to unity, if found convenient; for this does not fix the value of a variable in M, it only means that the M's remaining in the equations represent, not their values in pounds or tons, but their respective ratios to the absolute value of the particular M fixed to unity. Similarly any one of the $a$'s may be considered unity.

*Example.*—How many variables are there in a four-crank engine?

The answer is—

$$3\,(4 - 1) = 9.$$

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They are, or may be, rather—

$$\begin{aligned} \frac{M_2}{M_1}, \frac{M_3}{M_1}, \frac{M_4}{M_1} &= 3 \\ \frac{a_1}{a_2}, \frac{a_3}{a_2}, \frac{a_4}{a_2} &= 3 \\ \theta_{12}, \theta_{13}, \theta_{14} &= 3 \\ \hline &\quad 9 \end{aligned}$$

The double subscript to the $\theta$ indicates the particular numbers of the cranks between which $\theta$ is measured.

## 14. On the Selection of the conditional Equations.

If the eight conditional equations of Article 11 are satisfied, the solution is independent of the position of the reference plane. The possibilities of balancing an engine in which the number of variables is less than eight are to be investigated by solving a set of equations selected from the fundamental Group D, Article 11, equal in number to the number of variables concerned in the problem. For instance, the number of variables in a three-crank engine, whose cylinder centre lines are fixed, is $2(n - 1) = 4$. This indicates that four of the eight conditions may be satisfied.

The selection cannot be made at will. In the first place, the equations must be taken in pairs, that is, (1) and (2), (3) and (4), &c., must be taken together; and, in the second place, the solution of the chosen equations must be independent of the position of the reference plane; for there is no axis which can be specified as the particular one about which the engine will tend to turn, consequently any solution which is dependent upon the position of the reference plane is illusory, and is of no practical value. The following Theorem shows how this second condition operates to restrict the selection.

### Theorem.

In order that the balancing of the secondary couples may be independent of the position of the reference plane, the conditions for the balancing of the secondary forces must be satisfied as well.

Let the functions—

$f(M\,x\,y) = 0$, represent equation (1) and (2) of Group D, Art. 11. (*a*)
$F(M\,a\,x\,y) = 0$, represent equation (3) and (4) of Group D, Art. 11. (*b*)
$f'(M\,x\,y) = 0$, represent equation (5) and (6) of Group D, Art. 11. (*c*)
$F'(M\,a\,x\,y) = 0$, represent equation (7) and (8) of Group D, Art. 11. (*d*)

Suppose the reference plane moved a distance $z$ from its original position, then the



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values of all the $a$'s change by this amount, becoming $a_1 + z$, $a_2 + z$, &c. The functions ($b$) and ($d$) become

$$\mathrm{F}\ \{\mathrm{M}\,(a + z)\,x\,y\} = \mathrm{F}\,(\mathrm{M}\,a\,x\,y) + z\,.f\,(\mathrm{M}\,x\,y). \qquad (e)$$
$$\mathrm{F}'\ \{\mathrm{M}\,(a + z)\,x\,y\} = \mathrm{F}'\,(\mathrm{M}\,a\,x\,y) + z\,.f'\,(\mathrm{M}\,x\,y). \qquad (f)$$

The condition, that the balancing may be independent of the position of the reference plane, is, that the functions ($e$) and ($f$) vanish, when the functions ($b$) and ($d$) vanish. The first term of each of the functions ($e$) and ($f$) is similar in form to the functions ($b$) and ($d$), which, by supposition, vanish, leaving—

$$z\,.f\,(\mathrm{M}\,x\,y) = 0,$$

and—

$$z\,.f'\,(\mathrm{M}\,x\,y) = 0.$$

The $z$'s divide out, leaving functions of the same form as ($a$) and ($c$), which must vanish to make ($e$) and ($f$) vanish. Hence the Theorem.

### Corollary 1.

It is evident, from equation ($e$), that the primary couples cannot be balanced independently of the reference plane, unless the primary forces are balanced as well; for, in order that the expression may vanish, each term must separately vanish.

### Corollary 2.

It is also clear, from equation ($e$), that, if the primary force and couple polygons close for any one position of the reference plane, they will close for all positions of it.

### Corollary 3.

The unbalanced primary couple is constant for all positions of the reference plane, if the primary force polygon close. For, suppose $\mathrm{F}\ (\mathrm{M}\,a\,x\,y) = \mathrm{A}$, then, from equation ($e$), if $f\,(\mathrm{M}\,x\,y) = 0$, $\mathrm{F}\ \{\mathrm{M}\,(a + z)\,(x\,y)\} = \mathrm{A}$, for all values of $z$.

### Corollary 4.

The unbalanced secondary couple is constant for all positions of the reference plane, only if the secondary force polygon close.

This is proved in a similar manner to Corollary 3, by using equation ($f$).

It follows, from the above Theorem, that, to obtain any useful practical result, equations (7) and (8) cannot be taken, without at the same time taking equations (5) and (6), and that equations (3) and (4) must be accompanied by equations (1) and (2).

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Returning now to the selection of the four equations which contain the possibilities of balancing a three-crank engine, whose cylinder centre lines are given, it is evident that, under the operation of the two conditions stated above, the selection can only be made in two ways. These are—equations 1, 2, 3, and 4, or 1, 2, 5, and 6.

Any other set of four would be inconsistent with the condition that the equations be taken in pairs, or with the above Theorem. This at once shows that a three-crank engine may presumably be balanced,

(1) For primary forces and primary couples, leaving the secondary forces and couples unbalanced.

(2) For primary forces and secondary forces, leaving the primary and secondary couples unbalanced.

The possibilities of balancing an engine of this type are confined to these two cases, and no arrangement of three cranks is possible whereby anything further can be done; though there remains the practical problem, in all such cases of partial balancing, how to select the data so that the errors left, of necessity, are as small as possible.

The principles of the foregoing Articles are illustrated in the rest of the paper, by applying them to several typical cases.

## 15. Application of the Method to One and Two Crank Engines.

Obviously, nothing can be done to balance a one-crank engine without adding another crank. This evident fact may be used to illustrate the use of the expression giving the number of conditions which may be satisfied; for, substituting the value 1 for $n$ in expression (1), Article 13, it becomes

$$3\ (1 - 1) = 0,$$

which means that none of the conditions of balance can be fulfilled.

Considering a two-crank engine, the expression becomes

$$3\ (2 - 1) = 3,$$

showing that three of the balancing conditions may be satisfied. The conditions of Article 14 limit the selection to equations (1) and (2), from the fundamental Group D, Article 11, because the selection must be made in pairs; therefore, two only can be taken, and (1) and (2) must be taken before (3) and (4), by Corollary 1, Article 14, and it is evidently no use taking (5) and (6) until (1) and (2) have been satisfied. Hence equations (1) and (2) express all that can be done in the way of balancing a two-crank engine.

Taking these two equations to two terms each, they will at once reduce to values

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of $x$ and $y$, showing that the cranks must lie in the same plane of rotation, exactly opposite one another, the masses at crank radius being equal.

These two cases are given, merely to illustrate the way in which the expression giving the number of variables may be used at the beginning of a problem, to rigidly define the limits between which a solution is possible.

### 16. Application to Three-Crank Engine.

The number of conditions which may presumably be satisfied is equal to the number of variables, and this is, by equation (1), Article 13,

$$3\,(3-1) = 6.$$

The conditions of Article 14 limit the selection to the first six equations of the fundamental Group D, Article 11.

Generality is not lost by putting $x_1 = 1$, $y_1 = 0$, and $a_1 = 0$. This only means that No. 1 crank is put in coincidence with the line of reference O X, and that the reference plane is taken at No. 1 crank, as shown in Fig. 5.

Substituting these values in the equations selected, and taking them to three terms, the conditions expressing the possibilities of balancing a three-crank engine become—

$$\begin{array}{llll}
M_1 + M_2 x_2 & + M_3 x_3 & = 0 & (1) \\
M_2 y_2 & + M_3 y_3 & = 0 & (2) \\
M_2 a_2 x_2 & + M_3 a_3 x_3 & = 0 & (3) \\
M_2 a_2 y_2 & + M_3 a_3 y_3 & = 0 & (4) \\
M_1 + M_2 (x^2_2 - y^2_2) & + M_3 (x^2_3 - y^2_3) & = 0 & (5) \\
M_2 x_2 y_2 & + M_3 x_3 y_3 & = 0 & (6)
\end{array}$$

Eliminating the M's from equations (2) and (6)—

$$y_2 x_3 y_3 = y_2 x_2 y_3.$$

From this, either $x_3 = x_2$, in which case $y_3 = y_2$ (Article 8), or $x_3$ is not equal to $x_2$, in which case the above equation requires that $y_2 = y_3 = 0$, an untenable solution, since this requires that $x_2 = x_3 = 1$, which is contrary to the second supposition.

Taking $x_2 = x_3$, and, therefore, $y_2 = y_3$, it follows, from equation (2), that

$$M_2 = M_3.$$

Equations (1) and (5), therefore, reduce to

$$\left.\begin{array}{ll}
M_1 + 2\,M_2 x_2 & = 0 \\
M_1 + 2\,M_2 (x^2_2 - y^2_2) & = 0
\end{array}\right\}$$

Eliminating the M's—

$$x^2_2 - y^2_2 = x_2.$$

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that is—

$$(2x_2 + 1)(x_2 - 1) = 0,$$

since—

$$y^2_2 = 1 - x^2_2.$$

From this—

$$x_2 = -\frac{1}{2} = x_3,$$

or—

$$x_2 = 1 = x_3,$$

this is untenable, since it involves

$$y_2 = y_3 = 0,$$

and this has been shown above to require that $x_2$ is not $= x_3$. The first value of $x_2$ is, therefore, the only tenable one.

The values of $y_2$ corresponding to this are—

$$y_2 = y_3 = \pm\sqrt{1 - \frac{1}{4}} = \pm\frac{\sqrt{3}}{2}.$$

Substituting the values of $x_2$ and $x_3$ in equation (1)—

$$M_1 = M_2 = M_3.$$

From equation (2) it is clear that $y_2$ is of opposite sign to $y_3$, since the M's must be positive. The crank directions are, therefore, completely specified by Table II., from which it appears that the cranks are mutually at 120°.

TABLE II.

| Crank. | $x$. | $y$. |
|---|---|---|
| No. 1 ... ... ... ... ... | 1 | 0 |
| No. 2 ... ... ... ... ... | $-\frac{1}{2}$ | $\frac{\sqrt{3}}{2}$ |
| No. 3 ... ... ... ... ... | $-\frac{1}{2}$ | $-\frac{\sqrt{3}}{2}$ |

Substituting these values, in equations (3) and (4), they become, remembering that the M's are all equal—

$$-\frac{1}{2}a_2 - \frac{1}{2}a_3 = 0,$$

$$\frac{\sqrt{3}}{2}a_2 - \frac{\sqrt{3}}{2}a_3 = 0,$$

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From which,

$$a_2 = 0,$$
$$a_3 = 0,$$

indicating that the planes of revolution must be coincident, as shown in Fig. 6, Plate XLIII. Three equal masses, therefore, disposed in a plane of revolution, so that their respective radii are mutually at 120°, are balanced for primary and secondary forces. The force triangle corresponding to this is, of course, equilateral.

If the planes of revolution are not coincident, couple errors, both primary and secondary, will be introduced to an extent depending upon the distances apart of the three planes. This is, in fact, the case with the three-crank engine, which has been used so much in marine work.

Given that the masses are equal, and that the crank angles are mutually at 120° such an engine would be completely balanced for primary and secondary forces; but there remains a large couple error. Mr. Mark Robinson and Captain Sankey, assuming this arrangement, investigated the force errors very carefully, and communicated their results to the Institution in 1895. By the graphic method they used, no errors could be detected; calculation from an exact formula, however, disclosed a force error of about one pound in a 300 H.P. engine, running at 350 revolutions per minute, the ratio of connecting-rod to crank being 4·77 to 1.

17. There is another way of looking at the three-crank engine problem which may be noticed. Suppose it given that the cylinder centre lines are fixed. Then the number of independent variables in the problem is

$$2\,(3 - 1) = 4.$$

This shows that only four conditions can be satisfied, that is, that only four of the eight equations of Article 11 can be chosen. Now, consistent with the conditions of Article 14, the selection can only be made in two ways, viz.—

Equations Nos. 1, 2, 3, and 4,

or—

Equations Nos. 1, 2, 5, and 6.

The latter set have been fully considered in the first part of the preceding article, and they are completely satisfied by an engine with equal masses, and cranks mutually at 120°.

The first set merely lead to a solution for complete balancing, supposing the motion to be simple harmonic. The way to obtain the solution of 1, 2, 5, and 6 is as follows, assuming the initial conditions to be the same as before, and as shown by Fig. 5, Plate XLIII.

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Eliminating the M's from equations (2) and (4) of the preceding article,

$$y_2 a_3 y_3 = y_2 a_2 y_3;$$

$a_2$ is, by supposition, not equal to $a_3$, therefore

$$y_2 = y_3 = 0,$$

and therefore—

$$x_2 = x_3 = \pm 1;$$

since—

$$x^2 + y^2 = 1.$$

Substituting these values in equation 3—

$$\pm M_2 a_2 \pm M_3 a_3 = 0.$$

If $a_2$ and $a_3$ are both positive, $x_2$ must be of opposite sign to $x_3$, in order to leave the above two quantities connected with a minus sign, since the M's are always positive.

Suppose $x_2$ negative and $x_3$ positive, then the conditions of balance are stated by the two equations,

$$M_1 + M_3 = M_2, \text{ from equation (1)}$$

and—

$$M_2 a_2 - M_3 a_3 = 0.$$

The $a$'s are both of the same sign here, and, therefore, the corresponding planes of revolution lie on the same side of the reference plane which is at crank No. 1. Also, since all the $y$'s are zero, the cranks all lie in the same axial plane. The solution is shown in Fig. 7, Plate XLIII.

If the two $x$'s were of the same sign, the corresponding $a$'s would, of necessity, be of opposite signs, and the two planes of revolution would be on opposite sides of the reference plane.

## 18. Summary.

Summarising, the general conditions of balance for a three-crank engine with fixed centre lines are—

(1) The three cranks must lie in the same axial plane.

(2) The masses in the two outer planes of rotation must be such that their moments, with respect to the plane of rotation between them, must be equal and opposite, and the sum of the two outer masses must be equal to the central mass.

## 19. Application to a Four-Crank Engine.

The number of variables is—

$$3(4 - 1) = 9.$$

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In this case, all the conditions expressed by the fundamental group of equations D, Article 11, may, presumably, be satisfied.

Take the reference plane at No. 4 crank so that $a_4 = 0$, and put No. 1 crank in coincidence with the reference line O X, so that $x_1 = 1$, and $y_1 = 0$. These initial conditions are similar to those shown by Fig. 5, except that the reference plane is now at No. 4 crank, which is not shown there. Substituting these values, and taking four terms, in each of the equations of Group D, Article 11, they become—

$$\begin{array}{llll}
M_1 & + M_2 x_2 & + M_3 x_3 & + M_4 x_4 \quad (1) \\
0 & + M_2 y_2 & + M_3 y_3 & + M_4 y_4 \quad (2) \\
M_1 a_1 & + M_2 a_2 x_2 & + M_3 a_3 x_3 & + 0 \quad (3) \\
0 & + M_2 a_2 y_2 & + M_3 a_3 y_3 & + 0 \quad (4) \\
M_1 & + M_2 (x^2_2 - y^2_2) & + M_3 (x^2_3 - y^2_3) & + M_4 (x^2_4 - y^2_4) \quad (5) \\
0 & + M_2 x_2 y_2 & + M_3 x_3 y_3 & + M_4 x_4 y_4 \quad (6) \\
M_1 a_1 & + M_2 a_2 (x^2_2 - y^2_2) & + M_3 a_3 (x^2_3 - y^2_3) & + 0 \quad (7) \\
0 & + M_2 a_2 x_2 y_2 & + M_3 a_3 x_3 y_3 & + 0 \quad (8)
\end{array}$$

Equations (3, 4, 7, 8), in M $a$ above, are precisely similar in form to equations (1, 2, 5, 6) of Article 16, in M. The solution of (3, 4, 7, 8), in M $a$, is, therefore, the same as the solution of (1, 2, 5, 6) of Article 16, in M. Hence—

$$M_1 a_1 = M_2 a_2 = M_3 a_3 = 1, \text{ say.} \qquad \text{A}$$

And the cranks are mutually at 120°. Substituting the values of the M's in terms of the $a$'s, and the values of the $x$'s and $y$'s from Table II., Article 16, in equations (1, 2, 5, and 6), they reduce to—

$$M_4 x_4 = -\frac{1}{a_1} + \frac{1}{2a_2} + \frac{1}{2a_3} = M_4 (x^2_4 - y^2_4) \qquad (9)$$

$$M_4 y_4 = -\frac{\sqrt{3}}{2a_2} + \frac{\sqrt{3}}{2a_3} = -2 M_4 x_4 y_4. \qquad (10)$$

From (9)—

$$x^2_4 - y^2_4 = x_4,$$

that is—

$$(2x_4 + 1)(x_4 - 1) = 0. \qquad (11)$$

From (10)—

$$-x_4 y_4 = \frac{y_4}{2} \qquad (12)$$

From (11)—

$$x_4 = 1,$$

and therefore—

$$y_4 = 0;$$

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or—

$$x_4 = -\frac{1}{2},$$

and therefore—

$$y = \pm \frac{\sqrt{3}}{2}.$$

Both these solutions satisfy also equation (12).

Considering the first solution, it shows that crank No. 4 is parallel to crank No. 1, and, since the right and left hand expressions of equation (10) vanish, $y_4$ being zero, $a_2$ must be equal to $a_3$. The second solution leads to the same result, in terms of different letters. Hence, from A,

$$M_2 = M_3 \qquad B$$

Substituting the values—

$$x_4 = 1, \quad a_2 = a_3,$$

in equation (9)—

$$M_4 = \frac{1}{a_2} - \frac{1}{a_1}. \qquad (13)$$

From this it is clear that $a_1$ must always be greater than $a_2$, in order to make $M_4$ positive.

The relation between the masses is deduced from this equation with the aid of A, above, from which—

$$M_2\, a_2 = M_1\, a_1 = 1.$$

Substitute these values in the numerators of the terms on the left of equation (13), then—

$$M_4 = \frac{M_2\, a_2}{a_2} - \frac{M_1\, a_1}{a_1}.$$

From which—

$$M_4 + M_1 = M_2 = M_3. \qquad (14)$$

Summarising, a completely balanced four-crank engine must satisfy the following conditions:—

(1) The four cranks must be arranged in three planes of revolution, two of the cranks being in the central plane.

(2) The two cranks in the central plane must be at 120° with one another: the two outside cranks must point in the same direction in the same axial plane, which plane is at 120° with each of the cranks in the central plane of rotation.

(3) The masses in the central plane must be equal.

(4) The masses in the outer planes must be such that their moments, with respect to the central plane of rotation, are equal and opposite, and their sum must be equal to one of the equal masses in the central plane.



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The arrangement is shown in Fig. 8, Plate XLIV., and, though impracticable to realise, is useful in the solution of other problems.

### 20. Example.

Let the ratio $\frac{a_1}{a_2}$ be 3. Measuring from No. 4 crank—

$$a_1 = 3,$$
$$a_2 = a_3 = 1.$$

If $M_1 a_1$ = unity, from A—

$$M_1 = \frac{1}{3},$$

from B—

$$M_2 = M_3 = 1,$$

from (13) or (14)—

$$M_4 = \frac{1}{a_2} - \frac{1}{a_1} = \frac{2}{3}.$$

### 21. Four-Crank Engine satisfying Six Conditions.

Conditions 1, 2, 3, 4, 5, 6 may, presumably, be satisfied, leaving three of the nine variables concerned in a four-crank engine to be fixed arbitrarily. The solution of this problem trigonometrically was the main feature of Herr Schlick's paper at the 1900 Spring Meeting.

Choose the line of reference so that it bisects the angle between No. 1 and No. 4 cranks, as O X (Fig. 10, Plate XLIV.), then, $x_1 = x_4$, and, therefore, $y_1 = -y_4$.

The three conditions which may conveniently be fixed are—

(1) That the line of reference shall also bisect the angle between cranks No. 2 and No. 3. This involves that—

$$x_2 = x_3,$$

and, therefore, $y_2$ is equal in magnitude, and opposite in sign to $y_3$.

(2) That the ratio $M_1 : M_4 = 1$.

(3) That the ratio $a_1 : a_2$ is given.

The first two equations from Group D, Article 11, become, under these conditions—

$$2\,x_1 M_1 + x_2 (M_2 + M_3) = 0 \qquad (1)$$
$$0 + y_2 (M_2 - M_3) = 0 \qquad (2)$$

Considering equation (2), $M_2$ must evidently equal $M_3$, since $y_2$ is not zero, therefore (1) becomes—

$$M_1 x_1 + M_2 x_2 = 0 \qquad (3)$$

Taking a reference plane to bisect the distance between planes Nos. 1 and 4, so that $a_1 = -a_4$, the third equation from Group D becomes—

$$M_2 x_2 (a_2 + a_3) = 0;$$

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whence—

$$a_2 = -a_3.$$

The consequences of the three assumptions detailed above are, therefore, that $M_2 = M_3$, and $a_2 = a_3$, and are of opposite sign.

The cylinder lines are, therefore, symmetrical, with reference to the central plane of the engine. Substituting the foregoing equalities in the first six equations of Group D, Article 11, they become—

$$\begin{array}{lll} M_1 x_1 & + M_2 x_2 & = 0. \quad \text{A} \\ M_1 a_1 y_1 & + M_2 a_2 y_2 & = 0. \quad \text{B} \\ M_1 (x^2_1 - y^2_1) & + M_2 (x^2_2 - y^2_2) & = 0. \quad \text{C} \end{array}$$

Eliminating the M's from A and C—

$$x_1 (x^2_2 - y^2_2) = x_2 (x^2_1 - y^2_1).$$

Introducing $1 - x^2$ for $y^2$, this at once reduces to

$$x_1 x_2 = -\frac{1}{2}. \qquad \text{E}$$

Eliminating the M's from A and B—

$$x_1 y_2 a_2 = x_2 y_1 a_1.$$

Squaring each side, and substituting $-\frac{1}{2x_1}$ for $x_2$, from E, this reduces to

$$x^4_1 (4 a^2_2) + x^2_1 (a^2_1 - a^2_2) - a^2_1 = 0. \qquad \text{F}$$

If

$$P = \frac{a^2_1 - a^2_2}{8 a^2_2}, \text{ and } Q^2 = \frac{a^2_1}{4 a^2_2},$$

$$x^2_1 = -P \pm \sqrt{P^2 + Q^2} = \frac{M_2}{2 M_1}, \text{ also from E and A.} \qquad \text{G}$$

$x_1$ can be calculated from this, when the ratio $a_1 : a_2$ is given. Then $x_2$ can be found from equation E. The corresponding values of the $y$'s are found from the relation—

$$y^2 = 1 - x^2,$$

their signs being determined from equations A, B, C, D. The ratio $M_1 : M_3$ is found from equation G.

The relation given in equation E, above, is stated in the form $\cos \frac{\alpha}{2} \cos \frac{\gamma}{2} = \frac{1}{2}$, in Herr Schlick's paper.

## 22. Example.

Given that $M_1 = M_4$, that the crank angles are symmetrical, and that the ratio $a_1 : a_2 = 6{\cdot}5 : 2$. Find the crank angles, and the ratio of the masses $M_2 : M_1$, so that the engine may be balanced for primary and secondary forces and primary couples.

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The data necessitate a symmetrical engine, as shown in Fig. 6, Plate XLIII., *i.e.*, $M_2 = M_3$, and, taking the reference plane at the centre, so that $a_1 = a_4$; $a_2 = a_3$, of necessity.

Calculate the values of the quantities $P^2$ and $Q^2$, and solve equation G, of Article 21.

$$P = \frac{6{\cdot}5^2 - 2^2}{8 \times 2^2} = 1{\cdot}195\,;$$

therefore—

$$P^2 = 1{\cdot}428\,;$$

$$Q^2 = \frac{6{\cdot}5^2}{4 \times 2^2} = 2{\cdot}643.$$

Equation G becomes—

$$x^2_1 = -1{\cdot}195 \pm \sqrt{1{\cdot}428 + 2{\cdot}643},$$

therefore—

$$x^2_1 = +\,{\cdot}823,$$

the negative value being untenable, since $x_1$ must be real. Therefore, retaining the + sign, $x_1 = +\,{\cdot}908$.

From equation E—

$$x_2 = -\frac{1}{2 \times {\cdot}908} = -\,{\cdot}551.$$

Then—

$$y_1 = \pm\sqrt{1 - {\cdot}908^2} = \pm\,{\cdot}421,$$
$$y_2 = \pm\sqrt{1 - {\cdot}551^2} = \pm\,{\cdot}834.$$

The three assumptions made include—

$y_1$ is equal in magnitude and opposite in sign to $y_4$.

$y_2$ is equal in magnitude and opposite in sign to $y_3$.

The individual signs of $y_1$ and $y_2$ are to be determined from the general equations.

From equation B—

$$M_1\, a_1\, y_1 = -\,M_2\, a_2\, y_2.$$

$M_1$ and $M_2$ are positive, $a_1$ and $a_2$ are of the same sign, therefore $y_1$ must be opposite in sign to $y_2$.

Arranging the results—

| | |
|---|---|
| $x_1 = +\,{\cdot}908.$ | $y_1 = +\,{\cdot}421$, giving the direction of No. 1 crank. |
| $x_2 = -\,{\cdot}551.$ | $y_2 = -\,{\cdot}834$, giving the direction of No. 2 crank. |
| $x_3 = -\,{\cdot}551.$ | $y_3 = +\,{\cdot}834$, giving the direction of No. 3 crank. |
| $x_4 = +\,{\cdot}908.$ | $y_4 = -\,{\cdot}421$, giving the direction of No. 4 crank. |

From equation A—

$$M_1 : M_2 = M_4 : M_3 = x_3 : x_1 = {\cdot}551 : {\cdot}908 = {\cdot}607 : 1.$$

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Figs. 9 and 10, Plate XLIV., show the centre lines and crank angles, set out in their proper relative positions. The crank angles, in degrees, are added as a matter of interest.

### 23. Five-Crank Engine.

The number of variables in this case is—

$$3(5-1) = 12.$$

All the conditions expressed in the eight fundamental equations of Article 11 may presumably be satisfied, therefore, leaving four of the variables concerned in the problem to be fixed arbitrarily. The solution of the problem may be derived from the general solution of the four-crank problem given in Article 19, without reference to the eight equations of Article 11.

Fig. 8 shows the disposition of four cranks and the corresponding masses for complete balance. All the eight equations of Article 11 are satisfied. A combination of two such systems would also satisfy all the conditions of balance, for each is in equilibrium; and the combination of two systems, each in equilibrium, must form a single system in equilibrium. Combine two four-crank systems, of the type shown in Fig. 8, to form a single system, in the way shown in Fig. 11, Plate XLV., one system being distinguished by the radii of its masses being shown in full lines, the masses forming the system being $M_1$, $M_2$, $m_1$, and $m_2$, the other system being formed of the masses $M_3$, $M_4$, $M_5$, and $m_6$, their radii being shown dotted. It will be observed that the two systems are placed co-axially, with their central double cranks in the same plane of revolution, which plane may conveniently be chosen for the reference plane.

The angular disposition of the two systems, relatively to one another, is to be such that the four cranks in the reference plane are mutually at 120°. This arrangement is only possible, if one crank of the central pair, belonging to one system, coincides with one crank of the central pair, belonging to the other system, as shown in the reference plane (Fig. 11). The set of four masses in the reference plane may now be divided into two groups:—

(1) A group of three masses, $m_1$, $m_2$, $m_6$, whose radii are mutually inclined at 120°; these are indicated by cross hatching in Fig. 11.

(2) A single mass lettered $M_3$, and shown black; this is drawn at a slightly greater radius, for the sake of clearness. It is, of course, really coincident in position with $m_2$.

If the shaded masses were equal in magnitude, they would form a system in equilibrium amongst themselves, both for the primary and the secondary forces they give rise to. Article 16 is a proof of this, or it may be proved by substituting these values of the masses and angles in equations (1, 2, 5, 6), of Group D, Article 11). They might,

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under these circumstances, be subtracted from the combined system, without interfering with its equilibrium. Let them be equal, therefore, and let them be subtracted from the combined system; there will be left, in the reference plane, the single mass $M_5$.

Now, the masses forming each of the original systems must satisfy the relation expressed in equation (14), Article 19, which is, considering one system, that the magnitude of each of the two central masses is equal to the sum of the magnitudes of the two outer masses; the two outer masses are always, of course, in the same axial plane as shown in Fig. 8. Therefore, for the combined system under discussion, the following two equations must hold, the first line referring to the system whose mass radii are shown by thick lines, the second to the system whose mass radii are shown dotted in Fig. 11 :—

$$M_1 + M_2 = m_1 = m_2\,;$$
$$M_3 + M_4 = m_2 = m_6.$$

But $m_1 = m_2 = m_3$, by supposition; therefore—

$$M_1 + M_2 = M_3 + M_4 = M_5 \qquad (1)$$

And from the conditions of balance in the original four-crank systems—

$$M_1 a_1 = M_2 a_2 \qquad (2)$$
$$M_3 a_3 = M_4 a_4. \qquad (3)$$

Equations (2) and (3) each express the relation given by expression A in Article 19.

From these three equations the values of the masses for given values of the $a$'s may be found in terms of $M_5$. Notice that, in this case, fixing the pitch of the cylinders is equivalent to fixing four of the twelve variables concerned in the problem, and that the remaining eight quantities must be found from the equations.

Assuming the pitch of the cylinders to be settled, the magnitudes of the several masses may be more explicitly stated as follows :—

From equations (1) and (2) above—

$$M_1 + M_2 = M_5 = 1, \text{ say}\,; \qquad (4)$$
$$M_1 a_1 = M_2 a_2. \qquad (5)$$

Eliminating $M_2$ from (4) and (5)—

$$\left.\begin{array}{l} \left.\begin{array}{l} M_1 = \dfrac{a_2}{a_1 + a_2} \\ M_2 = 1 - M_1 \end{array}\right\} \text{in same axial plane} \\ \text{Similarly—} \\ \left.\begin{array}{l} M_3 = \dfrac{a_4}{a_3 + a_4} \\ M_4 = 1 - M_3 \end{array}\right\} \text{in same axial plane} \end{array}\right\} \text{at } 120^\circ.$$

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*Example.*—Suppose the pitch of the cylinders to be such that $a_1 = 2$ ft., $a_2 = 3$ ft., $a_3 = 4$ ft., $a_4 = 5$ ft., each of these distances being measured from the plane of the central crank of the five. Substituting, in the above expressions, and taking $M_5 =$ unity, the masses must be proportional to the numbers placed below them in the following rows:—

$$M_1 : M_2 : M_3 : M_4 : M_5$$

$$\frac{3}{8} : \frac{2}{5} : \frac{5}{9} : \frac{4}{9} : 1.$$

The crank angles have the same sequence as those shown in Fig. 12, Plate XLV.

## 24. Six-Crank Engine.

The number of variables in this case is—

$$3\,(6 - 1) = 15.$$

A solution of this problem may be derived from the general solution of the four-crank problem given in Article 19, by the method explained in the previous article, for a five-crank engine. Combine three four-crank systems of the type shown in Fig. 8, to form a single system, in the way shown in Fig. 13, Plate XLVI. The masses forming the systems being respectively—

$$\begin{array}{cccc} M_1 & M_2 & m_1 & m_2, \\ M_3 & M_4 & m_3 & m_4, \\ M_5 & M_6 & m_5 & m_6. \end{array}$$

It will be observed that the three systems are placed co-axially, with their central double cranks in the same plane of revolution, which plane may conveniently be chosen for the reference plane. The angular disposition of the three systems, relatively to one another, is to be such that the six cranks in the reference plane are mutually at 120°. This necessitates the arrangement shown in Fig. 13, in which the cranks form coincident pairs. The set of six masses in the reference plane may now be divided into two groups—

(1) A group of three masses $m_1$, $m_2$, $m_6$, shown cross-hatched, whose radii are mutually at 120°.

(2) A group of three masses, $m_{m_3}$, $m_4$, $m_5$, shown slightly displaced behind the former three, whose radii are mutually inclined at 120°.

If the masses in each group were equal, they would form two systems, each in equilibrium, both for the primary and the secondary forces they give rise to when reciprocated. (Article 16 contains the proof of this.) The two systems might, under these circumstances, be subtracted from the combined system, without interfering with

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its equilibrium. Let them be equal, therefore, and let them be subtracted from the combined system; there will be left six masses, viz., $M_1$, $M_2$, $M_3$, $M_4$, $M_5$, $M_6$, three on each side of the reference plane, forming a six-crank engine, whose cranks are shown in Fig. 14, Plate XLVI. The following equations are true of each system, respectively, from the conditions of balance of the original systems—

$$\begin{aligned} M_1 + M_2 &= m_1 = m_2, \\ M_3 + M_4 &= m_3 = m_4, \\ M_5 + M_6 &= m_5 = m_6. \end{aligned}$$

But $m_1 = m_2 = m_6$, and $m_3 = m_{m_4} = m_{m_5}$, by supposition; therefore—

$$M_1 + M_2 = M_3 + M_4 = M_5 + M_6; \qquad (1)$$

and, from the conditions of balance of the three original four-crank systems used in the combination,

$$M_1\, a_1 = M_2\, a_2, \qquad (2)$$
$$M_3\, a_3 = M_4\, a_4, \qquad (3)$$
$$M_5\, a_5 = M_6\, a_6. \qquad (4)$$

Equations (2), (3), and (4), severally, express the relations, A, in Article 19.

This solution adjusts itself most happily to the general conditions of design. The rules for designing a six-crank engine of this type may be thus stated:—

Take a reference plane, and group three pairs of cranks about it, each pair to be in an axial plane and to satisfy the conditions—

$$M_1\, a_1 + M_2\, a_2 = 0 \qquad (1)$$

$$M_1 + M_2 = a \text{ constant} = 1, \text{ say.} \qquad (2)$$

Assuming the spacing of the cylinders to be settled, all the $a$'s are known, and the magnitudes of the several masses may be more explicitly stated as follows:—

$$\left.\begin{array}{l} \left.\begin{array}{l} M_1 = \dfrac{a_2}{a_1 + a_2} \\ M_2 = 1 - M_1 \end{array}\right\} \text{in same axial plane} \\ \left.\begin{array}{l} M_3 = \dfrac{a_4}{a_3 + a_4} \\ M_4 = 1 - M_3 \end{array}\right\} \text{in same axial plane} \\ \left.\begin{array}{l} M_5 = \dfrac{a_6}{a_5 + a_6} \\ M_6 = 1 - M_5 \end{array}\right\} \text{in same axial plane} \end{array}\right\} \text{at } 120^\circ.$$

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25. Extension of the General Principles of the Foregoing Articles to the Balancing of Engines when the fundamental Expression 2, Article 2, includes Terms of higher Order than the Second.

If the extraction of the square root of expression ($a$), Article 2, be continued to more terms, the final expression for the displacement of B (Fig. 1) will contain terms in $4 \cos (\theta + \alpha)$, $6 \cos (\theta + \alpha)$, &c. This converging series of cosines, differentiated twice, gives a converging series for the acceleration of B (Fig. 1), which, when multiplied by the mass, gives the values of the unbalanced force acting on the frame. Expression (2), Article 2, then becomes—

$$M \omega^2 r \left[\cos (\theta + \alpha) + A \cos 2 (\theta + \alpha) - B \cos 4 (\theta + \alpha) + C \cos 6 (\theta + \alpha) \ldots\right], \qquad (1)$$

where the co-efficients A, B, C have the values, $c$ being the ratio $r/l$—

$$\begin{aligned} A &= c + 1/4\, c^3 + 15/128\, c^5 + \\ B &= \phantom{c + {}} 1/4\, c^3 + 3/16\, c^5 + \\ C &= \phantom{c + 1/4\, c^3 + {}} 9/128\, c^5 + \end{aligned}$$

The details of the calculation of the above expression are given in full in an interesting paper by Mr. John H. Macalpine, *Engineering*, October 22, 1897.

Each term in the series may be interpreted in the way explained in Article 4 for $\cos 2 (\theta + \alpha)$. Thus, the force corresponding to the third term may be considered as the result of the rotation of an imaginary crank, revolving four times as fast as the main crank. The fourth term represents an imaginary crank, revolving six times as fast, and so on.

For balancing purposes, therefore, the main crank shaft of an engine may be looked upon as associated with a series of imaginary crank shafts, rotating with speeds twice, four times, six times, &c., the speed of the main shaft, about the same axis of rotation; the angles between the cranks of the imaginary shafts being respectively twice, four times, six times, &c., the actual angles between the cranks of the main shaft, the masses carried by each imaginary shaft being in the same proportion as those of the main shaft, the planes of rotation of the series of imaginary cranks, corresponding to any one of the actual cranks, being, of course, coincident. Keyed to each shaft is an appropriate reference plane. The conditions that each imaginary shaft may be in balance are, simply, that force and the couple polygons belonging to it shall close. Consider a force vector belonging to one of the main cranks, and let its direction, with the initial line in the reference plane, be $\alpha$. The series of imaginary cranks belonging to this crank will have directions $2\alpha$, $4\alpha$, $6\alpha$, &c., with the initial lines in their respective reference planes.

These directions may be severally represented analytically in terms of the quantities specifying the direction of the force vector in the reference plane corre-



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sponding to the main crank. The way of doing this has been explained in Article 9 for the $2a$ direction. By an obvious extension of the method, a vector in the reference plane corresponding to the $4a$ crank is denoted by—

$$R\,\{(8y^4 - 8y^2 + 1) + i\,4xy\,(1 - 2y^2)\},$$

where R is a function of the mass carried by the main crank the length of the rod and the crank radius, but which, for balancing purposes, may be written = to M, since everything else cancels out when the final conditional equations are formed.

Similarly, a vector in the next plane of the series is represented by—

$$M\,[\,\{\,(x^2 - y^2)^3 - 3\,(x^2 - y^2)\,\} + i\,(6\,xy - 8\,x^3\,y^3)].$$

The eight conditional equations of Article 11 will be increased by four for every additional imaginary crank shaft taken in the series. Thus, corresponding to the $4a$ crank shaft will be the four additional conditions—

$$\begin{aligned} \Sigma\,M\,(8\,y^4 - 8\,y^2 + 1) &= 0 \\ \Sigma\,M\,.\,xy\,(1 - 2\,y^2) &= 0 \\ \Sigma\,M\,a\,(8\,y^4 - 8\,y^2 + 1) &= 0 \\ \Sigma\,M\,a\,.\,xy\,(1 - 2\,y^2) &= 0 \end{aligned}$$

Similar sets of four more must be added to the conditions, if the $6a$ crank shaft is taken in, and so on.

The solution of the set of sixteen simultaneous equations, corresponding to the main shaft, and the first three imaginary crank shafts, presents formidable analytical difficulties. They may be used, however, without much trouble, to test the balancing of a proposed arrangement; and, by the introduction of proper co-efficients, the magnitude of the unbalanced force and couple corresponding to any crank shaft in the series may be found.

Reverting to the polygons in the series of reference planes, it will be evident that, if a given arrangement of engine is in perfect balance, corresponding to the closed force polygon in the first and second reference planes (see Fig. 2), there will be a closed polygon in the third plane, whose sides are proportional in magnitude to the sides of the force polygon in the first plane, but make four times the angle with their initial line that the sides of the first polygon make with its initial line. There will be a closed polygon in the next reference plane, the direction of whose sides with their initial line is six times the angle of the sides of the first polygon with its initial line. And so on through the whole series. There will, of course, be a similar series of couple polygons.

The application of this test to the arrangement shown in Fig. 8 discloses that, not only is the primary and secondary shaft in balance, but that the imaginary shaft

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corresponding to $4a$, is also in balance for forces. The $6a$ shaft is out of balance. The $4a$ shaft is also balanced for couples. Since the $6a$ forces are not balanced, the $6a$ couples are not balanced independently of the position of the reference planes. In general, the infinite series of imaginary shafts are in balance, with the exception of the $6a$ shaft, and all multiples of it. If the reference plane is taken at the centre, the whole infinite series of couples are in balance, but those belonging to the $(n\,6)\,a$ series appear again, if the reference plane is taken in any other position; this follows from an extension of the Theorem of Article 14. To prove these statements, take the reference planes at the centre (Fig. 8), and the lines of reference in them, to correspond with the direction of the two outer cranks. The force polygon, in the first plane, is the equilateral triangle (Fig. 15). In the second plane, corresponding to the $2a$ shaft, it is also an equilateral triangle (Fig. 16). Fig. 17 shows the triangle for the $4a$ shaft still closed. In the next plane, for the $6a$ shaft, it becomes the line of Fig. 18. Continuing in this way, the $8a$, $10a$ polygons close, opening into a line for $12a$. The couple polygon for all planes is a line returning on itself, since the angle of the two outer cranks with the initial line is in each case 0, and therefore all the multiples are nothing.

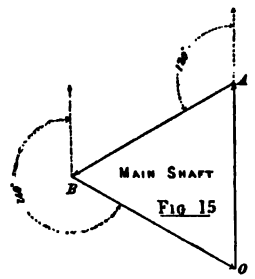

Fig. 15

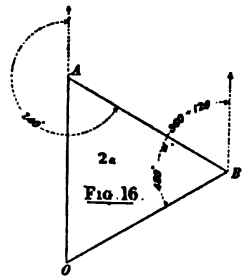

Fig. 16.

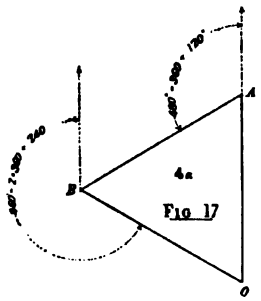

Fig. 17

Fig. 18

The actual magnitude of the maximum unbalanced force due to the $6a$ shaft is represented by O C (Fig. 18), and this is evidently $= 3$ O A.

O A represents to scale the maximum force due to the mass M. This is given by the value of—

$$\frac{M\omega^2 r}{g} \times C,$$

the fourth term in expression 1 of this article, so that it is only necessary to calculate the value of C. As a matter of interest, however, the whole series is given for the case of a rod equal to three and a half times the length of the crank. It is—

$$\frac{M\omega^2 r}{g}(\cos\theta + \cdot 291\cos 2\theta + \cdot 006\cos 4\theta + \cdot 0002\cos 6\theta + )$$

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The maximum unbalanced force is then—

$$\cdot 0006 \times \frac{M \omega^2 r}{g}$$

If $M = 5$ tons, $r = 2$ ft., revolutions per minute $= 88$, this force is ·0157 tons, which is equal to 33·6 lbs. weight. A quantity negligible compared with the unbalanced forces, even from the auxiliary engines about the ship.

From these properties of the arrangement of Fig. 8, it follows that the five and six-crank engines derived from it are similarly in balance. Thus, an engine balanced by the rules of Article 23, or a six-crank engine designed from Article 24, would be in balance for forces and couples up to those of the $6a$ class. There can be no doubt but that these are the kinds of engines to use to avoid vibration. The four-crank engine cannot approach them in this respect, since, even in the best arrangement, that of Article 21, couples of the $2a$ class of considerable magnitude are left unbalanced. Moreover, the crank angles of the five and six-crank engines fit in so well with the other conditions of design. A uniform crank effort diagram can be obtained with ease, and there are no awkward starting angles.

The general question of estimating the unbalanced primary and secondary force and couple, due to a stated arrangement of an engine, is briefly considered in the next Article. The different types being arranged in Table III., for convenience of comparison, amongst one another, and with the result just given for a five or six-crank engine.

## 26. Estimation of Errors.

To estimate the maximum values of the primary and secondary unbalanced forces and couples, due to a given set of crank angles and masses, it is only necessary to find the several magnitudes of the closing sides of the four polygons. For instance, in the secondary force polygon, the "closure," reversed, represents the vector sum, or resultant, of all the secondary forces, and its projection on the plane containing the cylinder centre lines is the instantaneous value of the secondary disturbing force, and this is a maximum, when the closure itself turns into the plane. The magnitude of the closure, reversed, for a particular polygon, therefore, represents the maximum error due to that unclosed polygon, and it is to be calculated, by substituting the given values of the M's, the $a$'s, and the $x$'s and $y$'s in the two equations representing the polygon. The one equation gives the sum of the projections of the several sides on the axis of X, represented by X, say; the other equation, the sum of their projections on the axis of Y, represented by Y, say. The magnitude of the closure itself is then given by $\sqrt{X^2 + Y^2}$. The following statement, giving the closures for each of the four polygons, is added for convenience of reference.

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Primary force polygon—

$$X = \frac{\omega^2 r}{g} \Sigma (M x); \qquad Y = \frac{\omega^2 r}{g} \Sigma (M y).$$

Magnitude of the closure is—

$$\sqrt{X^2 + Y^2}.$$

Primary couple polygon—

$$X = \frac{\omega^2 r}{g} \Sigma (M a x); \qquad Y = \frac{\omega^2 r}{g} \Sigma (M a y).$$

Magnitude of the closure is—

$$\sqrt{X^2 + Y^2}.$$

Secondary force polygon—

$$X = \frac{\omega^2 r^2}{g l} \Sigma \{M (2 x^2 - 1)\}; \qquad Y = \frac{\omega^2 r^2}{g l} \Sigma \{ M (2 x y)\}.$$

Magnitude of the closure is—

$$\sqrt{X^2 + Y^2}.$$

Secondary couple polygon—

$$X = \frac{\omega^2 r^2}{g l} \Sigma \{M a (2 x^2 - 1)\}; \qquad Y = \frac{\omega^2 r^2}{g l} \Sigma \{M a (2 x y)\}.$$

Magnitude of the closure is—

$$\sqrt{X^2 + Y^2}.$$

The direction of a closure, reversed, relatively to the axis in the reference plane, is $\tan^{-1} \frac{Y}{X} = \alpha$.

In particular cases the expressions for the several magnitudes of these closures assume simple forms, illustrations of which are given in the following types:—

Let A represent $\frac{\omega^2 r}{g}$, and B $\frac{\omega^2 r^2}{g l}$.

*Type* 1.—Three-crank engine, the cranks all being in the same plane, the masses being proportioned by Article 17, so that the primary force and couple polygons are closed, but the secondary force and couple polygons are open to an unknown extent. The values to be used in the general equations of Article 25 are, taking the reference plane at the central crank—

$$x_1 = -x_2 = x_3 = 1,$$
$$y_1 = \quad y_2 = y_3 = 0,$$
$$M_1 a_1 = M_3 a_3 \quad \text{and} \quad M_1 + M_3 = M_2.$$

The closure for the secondary force polygon reduces to—

$$2 B M_1 \left(\frac{a_1 + a_3}{a_2}\right),$$

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in terms of the pitch of the cylinders; or—

$$2\,B\,(M_1 + M_3) \quad \text{or} \quad 2\,B\,M_2,$$

in terms of the masses.

The length of the closure to the secondary couple polygon depends upon the position of the reference plane, since there is secondary force error (Theorem 1, Article 14).

If the reference plane is taken at the central crank, the secondary couple polygon closes, and there is, therefore, no error; if it be taken at an outer crank, the length of the closure is 2 B M $a$, the M and $a$ being respectively the mass and the distance, corresponding to the crank farthest from the reference plane.

*Type* 2.—Three-crank engine, arranged as in Article 18.

Taking the reference plane at the central crank, the quantities to be substituted in the general equations, Article 25, are—

$$x_1 = 1, \qquad x_2 = -\frac{1}{2}, \qquad x_3 = -\frac{1}{2},$$

$$y_1 = 0, \qquad y_2 = \frac{\sqrt{3}}{2}, \qquad y_3 = -\frac{\sqrt{3}}{2},$$

$$M_1 = M_2 = M_3.$$

In this case the primary and secondary force polygons are both closed, the remaining two polygons being open.

The closure for the primary couple polygon reduces to—

$$A\,M\,\sqrt{a^2_1 + a^2_3 + a_1\,a_3},$$

and for the secondary couple polygon to—

$$B\,M\,\sqrt{a^2_1 + a^2_3 + a_1\,a_3}.$$

If the cylinders are equally spaced, so that $a_1 = a_3$, the length of the two closures are, respectively—

$$A\,M\,a\,\sqrt{3} \quad \text{and} \quad B\,M\,a\,\sqrt{3}.$$

The magnitudes of both the primary and secondary couple errors are constant, because both the primary and secondary force polygons close.

*Type* 3.—Symmetrical four-crank engine, arranged as in Article 21, and illustrated by Figs. 9 and 10.

The general data, in this case, are, the reference plane being at the centre—

$$x_1 = \quad x_4, \qquad -x_2 = -x_3,$$
$$y_1 = -y_4, \qquad y_3 = -y_2,$$
$$a_1 = -a_4, \quad \text{and} \quad a_2 = -a_3.$$

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The only open polygon is that representing the secondary couples; the other three are closed, by the conditions of the problem. The length of the closure is—

$$4\,B\,(M_1\,a_1\,x_1\,y_1 + M_2\,a_2\,x_2\,y_2).$$

Referring to equation G, Article 21, it will be noticed that—

$$x^2_1 = \frac{M_2}{2\,M_1}.$$

By means of this relation, and

$$a_2 = -\frac{M_1\,a_1\,y_1}{M_2\,y_2}, \text{ from equation B, Article 21,}$$

and

$$x_2 = -\frac{1}{2\,x_1}, \text{ from equation E,}$$

the $x$'s and $y$'s may be eliminated from the above expression for the error; for, substituting these values for $x_2$, $a_2$, and putting $\sqrt{1 - x^2_1}$ for $y_1$, and multiplying and dividing by $x_1$, where necessary, to obtain the $x^2_1$ form, the expression becomes—

$$2\,B\,a_1 \left\{ M_1 \sqrt{\frac{1}{x^2_1} - 1} + M_2 \sqrt{\frac{1}{x^2_1} - 1} \right\};$$

and this, substituting $\frac{M_2}{2\,M_1}$ for $x^2_1$ gives—

$$2\,B\,a_1 \left\{ (M_1 + M_2) \sqrt{\frac{2\,M_1}{M_2} - 1} \right\},$$

an expression corresponding to equation (52) in Herr Schlick's paper.

*Type* 4.—Four-crank engine with cranks at 90°, and equal reciprocating masses.

Take the reference plane at the centre. The data are—

$$\begin{aligned} x_1 &= +1, & y_1 &= 0, \\ x_2 &= -1, & y_2 &= 0, \\ x_3 &= 0, & y_3 &= +1, \\ x_4 &= 0, & y_4 &= -1. \end{aligned}$$

$$M_1 = M_2 = M_3 = M_4, \quad \text{and} \quad a_1 = -a_4 \quad \text{and} \quad a_2 = -a_3.$$

Substitute these values in the proper equations, and the following results will be obtained:—

Primary force polygon,

$$\text{Length of closure} = 0.$$

Primary couple polygon,

$$\text{Closure} = M \sqrt{(a_1 - a_2)^2 + (a_4 - a_3)^2},$$

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which is equal to—

$$M(a_1 - a_2)\sqrt{2},$$

if the distances between each pair of cranks, which are at 180°, are equal.

The secondary force polygon reduces to 2 M B, and the secondary couple polygon to M B $(a + a_2)$. Since there is secondary force error, the secondary couple error depends upon the position of the reference plane.

## 27. Comparative Examples.

It is interesting and instructive to compare the disturbing effect, due to different arrangements of an engine, on the assumption that the least reciprocating masses are equal in each case, and that the engine cylinders are spaced a given distance apart. For this purpose, assume the following data:—

Mass of the lightest set of reciprocating parts = 5 tons.

Crank radius = 2 ft. Cylinders 16 ft. pitch.

Ratio of crank to rod = 1 : 3·5.

Revolutions per minute = 88.

Then—

$$\omega = 9{\cdot}2 \text{ approximately.}$$

$$\omega^2 = 84{\cdot}5.$$

$$\frac{\omega^2 r}{g} = 5{\cdot}25 = \mathbf{A}.$$

$$\frac{\omega^2 r^2}{g\,l} = 1{\cdot}5 = \mathbf{B}.$$

The force errors may be compared with the maximum disturbing force, due to the reciprocation of the lightest mass, with simple harmonic motion, which, in the case under discussion, is $\frac{5\,\omega^2 r}{g} = 26{\cdot}25$ tons.

The different results are set forth in Table III. The general formulæ for the errors will be found in the first horizontal row, corresponding to each type. The second row gives the forms the formulæ assume when the pitch of the cylinders is equal. The figures in the third row are the numerical results corresponding to the above data.

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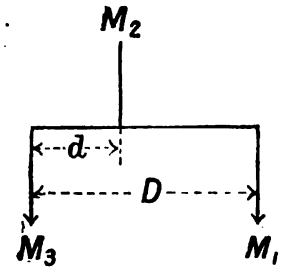
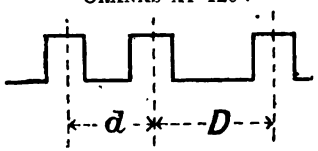
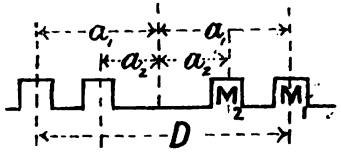
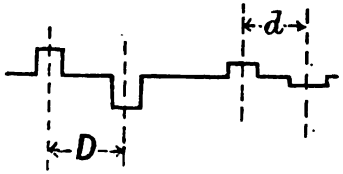
TABLE III.

| Type. | Primary Force Error. Tons. | Primary Couple Error. Foot Tons. | Secondary Force Error. Tons. | Secondary Couple Error. Foot Tons. |
|---|---|---|---|---|
|   | 0 | 0 | $2\,B\,M_1 \frac{D}{d}$ or $B\,2\,(M_1 + M_3)$<br>$4\,B\,M_1$<br>30 | Variable.<br>Reference plane at centre, error = 0, at number 1 = 480. |
| CRANKS AT 120°.   | 0 | $A\,M\,\sqrt{D^2 + d^2 + D\,d}$<br>$A\,M\,D\,\sqrt{3}$<br>728 | 0 | $B\,M\,\sqrt{D^2 + d^2 + D\,d}$.<br>$B\,M\,D\,\sqrt{3}$.<br>208 |
| SYMMETRICAL ARRANGEMENT.   | 0 | 0 | 0 | $B\,(M_1\,a_1\,x_1\,y_1 + M_2\,a_2\,x_2\,y_2)$.<br>$B\,D\left\{(M_1 + M_2).\sqrt{\frac{2\,M_1}{M_2} - 1}\right\}$<br>$1{\cdot}28\,B\,M_1\,D$.<br>465 |
| CRANKS IN 180°, PAIRS AT 90°.   | 0 | $A\,M\,\sqrt{D^2 + d^2}$<br>$A\,M\,D\,\sqrt{2}$<br>594 | $2\,M\,B$<br>$2\,M\,B$<br>15 | $2\,M\,B\,D$.<br>Reference plane at centre<br>240 |

Revolutions per minute, 88. $A \doteq 5{\cdot}25$. $B = 1{\cdot}5$. Disturbing force due to the lightest mass of 5 tons is 26·25 tons.



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## DISCUSSION.

Mr. JOHN H. MACALPINE (Member): My Lord and Gentlemen, before offering the remarks I have prepared bearing on the subject of this important paper, I wish to express the indebtedness we must all feel to Professor Dalby for the magnificent way in which he has delivered his lecture. He has made a difficult subject luminous by his excellent models and lantern slides, and I have no doubt we have all now a much clearer grasp of the whole question. This Institution is to be congratulated on the fact that from its Transactions alone almost the whole history of the growth of thought on this important and fascinating question of steamship vibration could be written. We are greatly indebted to the conscientious and painstaking labours of Messrs. Yarrow, Schlick, Macfarlane Gray, Mallock, Professor Dalby, and others for knowledge we now possess. The papers we have listened to to-day contribute materially to deepen that knowledge. I wish to direct attention only to a few of the many questions these papers suggest. First, the whole question is a recognition of the elasticity of the ship; but, in all calculations that have been made for balancing the forces from the various cranks, the fundamental assumption is that, for the length of the engine, the forces act on a rigid structure. If I apply with three fingers a system of balanced periodic forces to a stiff ruler I produce no sensible flexure or vibration; but if I treat in the same way a thin wand the flexure or vibration is obvious. Hence, if we balance an engine for first period forces, by the interaction of the forces from several cranks, we can only hope that the ship will be approximately free from vibrations of this period. I think experience fully confirms this expectation of only partial freedom; especially in the case of full-powered ships, where the length of the engine is a considerable fraction of the length of the ship. This is an effect, the importance of which has always forced itself on my attention when thinking and writing on the present subject; and I think it has been too much overlooked. The increase in the number of cranks advocated by Professor Dalby would, by lengthening the engine, generally lead to less satisfactory results on this account; except in such arrangements as that of the six-crank engines of the *General Baquedano*, where cranks applying opposite forces of first period have been placed together, no doubt to diminish the effect of elasticity. I think it is also obvious that, for higher period forces, calculations for balance on the assumption of rigidity are less admissible than for first period, as the nodes for the former are closer than those for the latter. Hence the wisdom of the advocacy of five, six, or more crank engines to annul these high-period vibrations is doubtful. On account of expense and extra complication it is also to be avoided. Secondly, in considering short-period vibrations, it is fundamentally inaccurate to look only to the smallness of the exciting forces. As is well known, as Herr Schlick has noted, and as I have pointed out on more than one occasion in writing on steamship vibration, when there is sychronism, the epoch of the ship vibration falls one quarter period behind that of the exciting force, and there is thus produced a continual flow of energy to the ship. It is only when the absorption of energy of the ship vibration due to resistance of the water, viscosity of the material of the ship, &c., equals this communication of energy, that the increase of amplitude of any vibration ceases. The question is well illustrated by the case of a ship rolling in a set of synchronous waves, where a very large angle of roll may be induced by an almost imperceptible swell. Again, taking the co-efficient of viscosity from Tomlinson's experiments on the torsion of a steel wire, I have shown in an

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investigation on "The Inertia Stress of Elastic Gears," Chapter XIII., published in the last volume of the "Journal of the American Society of Engineers," that when viscosity alone acts, the amplitude of the synchronising vibration will be increased to between 1,500 and 1,600 times the amplitude of the exciting vibration. No doubt in ship vibration the effects are more moderate on account of more powerful absorption of energy, but they are still large. In short, these high period vibrations and tremors do present themselves, and are often a source of great discomfort. To see them we have only to look at the excellent vibration diagrams Herr Schlick has shown us to-day. They must be dealt with in any complete solution of the vibration problem. They have to-day been ascribed by Herr Schlick much more exclusively to the action of the propeller than, I think, the facts warrant. Thirdly, in Article 15, Professor Dalby states:—"Obviously, nothing can be done to balance a one crank-engine." On the contrary, I think it will only take a moment to show that no complete solution of the balancing problem can be attained, unless we balance each crank separately; and that, if we do this, a solution both complete and highly practical can be obtained. In my paper published in *Engineering*, October, 1897, to which Professor Dalby has so kindly referred, on the "Analysis of the Inertia Forces of the Moving Parts of an Engine," I had intended to add a discussion of the possibilities of different numbers of cranks, such as Professor Dalby has now given us. After writing down a similar statement to that just quoted, it struck me, on again looking at the formula for inertia force, that the possibilities of the one-crank engine had not been exhausted. The formula is—

$$F = \frac{M R \omega^2}{g} (\cos\theta + a \cos 2\theta + b \cos 4\theta + \&c.),$$

where $a$ and $b$ are numerical co-efficients readily calculated, M the moving mass, R the crank radius, and $\omega$ the angular velocity of the crank. With Mr. Yarrow's bob-weights, or Herr Schlick's four-crank engine, we can annul the effect of the first term, $\cos\theta$, of the second factor—always supposing the ship rigid under the engines. Subject to the same limitation, we can, by the five and six-crank engine, annul the effect of the three terms $\cos\theta$, $a \cos 2\theta$, and $b \cos 4\theta$; and by my first patent we can deal with any term or terms which happen to be giving trouble. Greater numbers of cranks have, no doubt, greater possibilities. But the series is infinite, and the number of cranks must be finite. But the expression for F has the factor $\frac{M R \omega^2}{g}$; and if from the same crank we actuate another mass, or masses, giving an inertia force

$$-\frac{m r \omega^2}{g} (\cos\theta + a \cos 2\theta + b \cos 4\theta + \&c.),$$

it is only necessary to make

$$M R = m r$$

to obtain $F = 0$. And in no other way can F be made zero. The solution of the balancing problem is unique. The diagram on the wall* gives a form of engine most convenient for marine purposes, in

* The diagram to which reference is here made illustrated the engine designed by Mr. Macalpine. A full description and illustrations of this engine will be found in the account of the Proceedings at the Glasgow Meeting, on a subsequent page of this volume.—ED.

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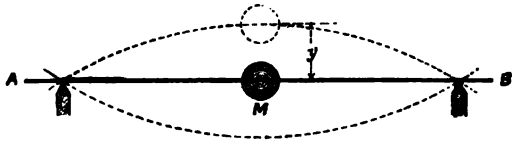
which this is realised. The horizontal and vertical forces, arising from the crank and a properly determined portion of the connecting rod, are exactly balanced by revolving counterbalance weights. The rest of the connecting rod, together with the other reciprocating parts over the crank, are equal to those in line with the cylinder not above the crank, supposing equal stroke for the two cylinders. The horizontal forces from the two ends of the levers are equal. The valve gears—one to each pair of cylinders—are similarly balanced. Thus the horizontal and vertical balance is perfect. But the engine is unbalanced torsionally. However, it can readily be shown, that the slowest torsional vibrations which will be set up by engines near the centre of a ship—those with two nodes—are many times as fast as the speed of revolution of the engine which would be put in the ship. For instance, a ship 400 ft. long would vibrate torsionally about 1,000 times per minute. If the engine ran, as it probably would, from 100 to 120 revolutions per minute the torsional vibrations are about the tenth or eighth period. And it can further be shown that the torsional couples in the proposed engine for the fourth, sixth, eighth, or higher periods are not sensibly greater than in an engine with the cylinders both over the crank. Hence, as elastic torsional vibrations do not arise from an ordinary engine, they will not arise here. The torsional couple of first period is large. Since there is no synchronism, it will produce an easily calculated rigid body vibratory rotation of the ship. In submitting, at his own request, the whole investigations, of which I am here only stating some results, to my friend Admiral Geo. W. Melville, Engineer-in-Chief of the U.S. Navy, who, after careful examination, has approved the design, I calculated this effect for the U.S.S. *Alabama*, supposing engines of the same power and stroke as she has. The effect at the extreme outside, 36 ft. from the centre, was a vibration of less than $\frac{1}{200}$ in. amplitude. That is, a total movement of less than $\frac{1}{100}$ of an inch; so that, training a gun broadside on an object three miles distant, the sight would appear to have a movement on the target of $4\frac{1}{2}$ in., which would be quite invisible. The effect is independent of the speed of revolution of the engines; and the similar effects from the second and higher period couples are relatively insignificant. To make the lever connection satisfactory is only a question of giving sufficient strength and stiffness and ample bearing surfaces. The engine is quite accessible. It is also lighter and much more compact than the ordinary design, and all question of rigidity of the ship under the engine is eliminated. I have had to say so much in justification of my statement that the engine is a complete solution of the balancing problem. I trust I have kept within the legitimate bounds of the present discussion. Possibly, at the next meeting, I may be honoured by being allowed to give the reasons for my present statements.

Professor W. E. Dalby, M.A., B.Sc. (Associate): It is certainly the fact that up to the present authors of the various papers on balancing engines, to be found in the Transactions of the Institution, make the tacit assumption that the engine frame is rigid, or that the local deflections of the frame are negligible as causes tending to set up vibrations of the hull. The engine frame could probably be sufficiently stiffened against local deflection if it were found to be yielding enough to cause vibration. It is true that a very small periodic force acting upon an elastic system is capable of forcing vibrations of relatively enormous magnitude at a sychronising speed, but there are not enough data available to calculate the amplitude of the forced vibration in the neighbourhood of synchronising speeds with any approach to accuracy in the case of a ship's hull. The amplitude of the vibration is limited by the forces arising from imperfect elasticity friction between the sides of the vessel and the water, &c., acting to quell the motion, and, unless

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these are known, any calculation of the amplitude of the forced vibration is meaningless. To illustrate this point, let the steel bar A B support a mass M at its centre, and let the system make naturally 286·48 vibrations per minute. Let $\mu$ be the force required to hold M at unit displacement. Suppose, further, that the mass contains an engine, in which there is an unbalanced force E cos $p\ t$ acting vertically, $P = 2\pi n$, $n$ being the revolutions per second, and that the frictional resistance to the motion of M varies directly as the velocity. Let $y$ be the displacement of the centre of M from its mean position, $\dot{y}$, M's velocity, and $\ddot{y}$ its acceleration. Then, if the mass is displaced a distance, the equation of motion of the system takes the well-known form, neglecting the mass of the bar—

$$M\ddot{y} + \delta\dot{y} + \mu y = E\cos p\,t, \text{ *}$$

of which the solution is—

$$y = \frac{P\sin\epsilon}{p\,b}\cos(p\,t - \epsilon), \qquad (1)$$

where the value of $\epsilon$, which fixes the phase of the motion, is given by—

$$\tan\epsilon = \frac{p\,b}{q^2 - p^2}. \qquad (2)$$

In (1) and (2), $P = E/M$, $q^2 = \mu/M$, $b = \delta/M$.

Synchronism takes place when $q = p$. The phase difference is then 90°, and the maximum value of $y$ is given by—

$$y = \frac{P}{p\,b}.$$

In order to graphically represent the effect of varying the value of $b$ on the forced vibration produced in the neighbourhood of a synchronising speed, $p = 2\mu n$, $n$ being the revolutions per second of the engine, has been taken at gradually increasing values, through the synchronising speed of 286·48 revolutions per minute, and the value of the maximum displacement of $y$ calculated, P being unity. The results of these calculations are shown in the diagram on page 142. Supposing the engine to be running at 280 revolutions per minute, the maximum value of $y$ is practically the same for all values of $b$ taken, and quite insignificant in amount. In fact, the effect of varying $b$ is not felt until the engine runs at about 285 revolutions per minute, the maximum amplitude being still insignificant. At 286 the effect of the variation of $b$ is more marked, and at the synchronising speed its maximum influence is shown. Beyond this speed the curves rapidly drop towards insignificant values again. This, though only a hypothetical example of the simplest system, sufficiently indicates how necessary it is to include the resisting forces in any actual problem in order to get even an approximately true result. The phase difference corresponding to the different speeds, and for three of the different values of $b$, are shown in Figs. 3, 4, 5. O X in each corresponds to the position at which the force causing the oscillation is a maximum. The statement in the paper that "nothing can be done to balance a one-crank engine"—on which Mr. Macalpine founds so much of his criticism and illustration—continued, runs, "without the addition of another crank:" meaning since the paper was all about reciprocating masses without the addition of another reciprocating mass. Nevertheless,

* Vibrating systems of one degree of freedom can be studied in Lord Raleigh's "Sound." Special illustrations are given in Perry's "Calculus for Engineers."

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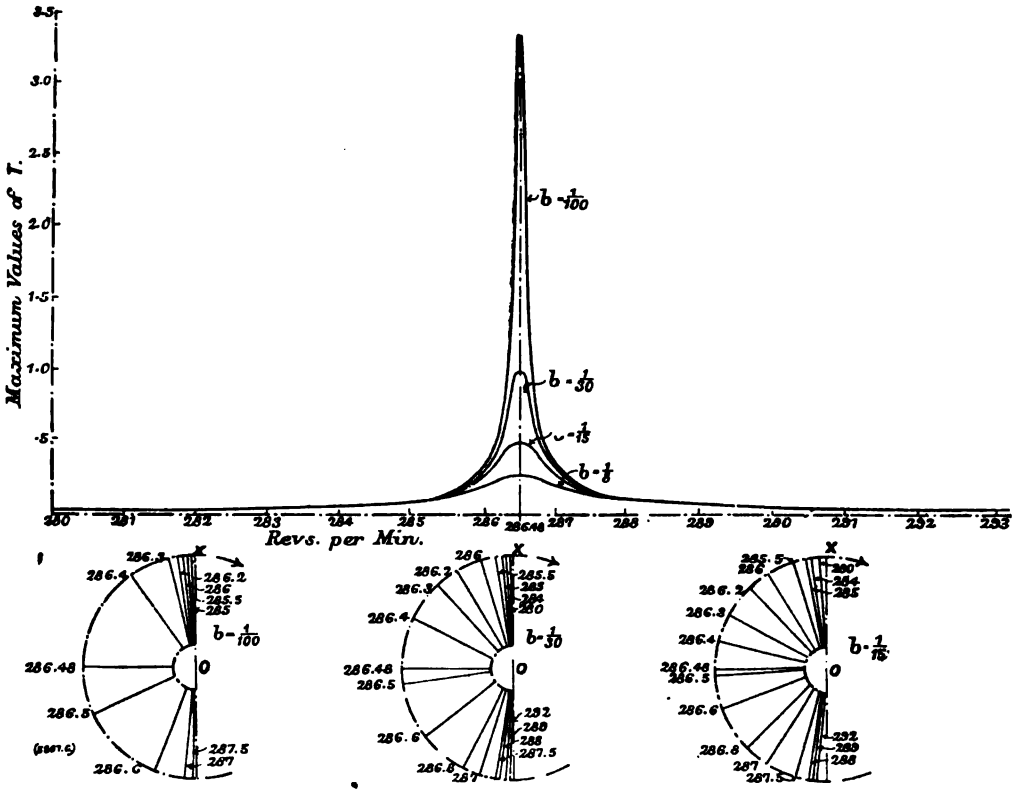
Mr. Macalpine's evident oversight in reading the sentence provided an opportunity for the description of an exceedingly interesting form of balanced engine, in which two cylinders placed athwart the




crank shaft are connected to one crank, the reciprocating masses being put in exactly opposite phase and in the same plane of revolution.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, I beg to move a hearty vote of thanks to Professor Dalby for his most interesting paper, and I am sure you will all join with me in that.

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# THE MOTION OF SUBMARINE BOATS IN THE VERTICAL PLANE.

By Commander WILLIAM HOVGAARD, Royal Danish Navy, Member.

[Read at the Spring Meetings of the Forty-Second Session of the Institution of Naval Architects, March 29, 1901; the Right Hon. the Earl of GLASGOW, G.C.M.G., LL.D., President, in the Chair.]

IN order that a submarine boat should be under perfect control in the vertical plane, it should, when going on the surface, be able to dive quickly to any desired depth of immersion. It should be able to keep that depth with certainty and facility within narrow limits, and should be capable of again emerging quickly to the surface at any time.

The boat must therefore possess the two qualities—Stability of Motion and Manœuvring Power. Each of these qualities is examined in the following pages, and also the manner in which they are affected by varying the rudders, the amount and distribution of buoyancy, the shape of hull, &c. But, in order to form a precise opinion on these somewhat complicated questions, it is necessary to treat the subject mathematically.

The mathematical investigation falls into two parts, steady motion and disturbed motion, each of which is treated separately, and the latter in its various forms.

In order to facilitate the mathematical treatment and the subsequent discussion of varying dispositions, it has been found advantageous to start with the following suppositions, which embody the simplest case:—

The boat is perfectly symmetrical about a vertical and a horizontal longitudinal plane, and is propelled by one screw, having its shaft in the intersection of these two planes—the axis of the boat.

There is one rudder, which is placed aft.

The buoyancy exceeds the weight of the boat by a small amount; the surplus buoyancy generally not exceeding one five-hundredth part of the displacement.

A conning tower is placed vertically above the centre of gravity, and the boat is balanced so that, when floating at rest on the surface its axis is horizontal, the top line of its hull just in, or parallel to, the water-line, and only the conning tower, or

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part of the conning tower, projecting above the surface. When the boat dives, the displacement of that part of the conning tower which projected above water when at rest will constitute the surplus buoyancy. The slight non-symmetry due to the conning tower is at first neglected.

## Steady Motion.

We shall first consider the important case of steady motion in a straight line on a certain depth of immersion.*

In order that steady motion may be possible, the centre of gravity should be fixed in position, the weight and buoyancy, as well as the speed of the boat, should be constant, and no external forces should be acting. On account of the symmetry of the boat, we need only consider forces lying in a vertical plane containing the axis.

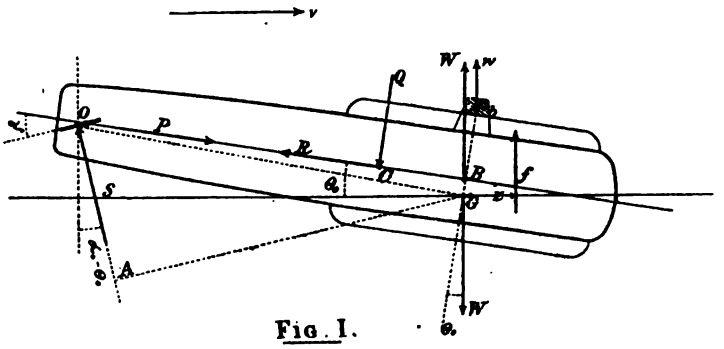

Fig. I.

Let the velocity of the boat, along its horizontal trajectory, be $v$.

In order to counteract the surplus buoyancy, the boat must be inclined downwards by the head, at a small angle, $\theta_0$, to the horizon. As the boat must, necessarily, possess stability, the rudder must be carried at a certain small angle, $\alpha_0$, downwards, in order to keep the boat thus inclined.

Both $\alpha_0$ and $\theta$ being small, we shall here, and in what follows, substitute the angles in circular measure for their sines, and put their cosines equal to 1.

The forces which act on the boat, and which, in steady motion, must be in equilibrium, are the following :—

The weight of the boat, W, acting vertically downwards through the centre of gravity, G, situated at a small distance, B G, below the centre of buoyancy.

The buoyancy of the boat, W, exclusive of the surplus buoyancy, acting vertically upwards through the centre of buoyancy, B, which, from symmetry, must be situated in the axis of the boat.

* This part of the problem has been formerly treated in a somewhat different way by a French Naval Architect, M. Leflaive, Ingénieur de la Marine, in "Memorial de Génie Maritime," 1896. Besides the question of steady motion in a horizontal line, M. Leflaive discusses also the case of small oscillations, but does not include any form of resistance, excepting that due to the horizontal translatory motion. His able work has formed a valuable point of issue for the following investigation, and should be studied by all who take an interest in this subject.

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The surplus buoyancy, $w$, acting vertically upwards, through the centre of buoyancy, $b$, of that part of the conning tower which projects above water when at rest.

The resistance of the water to the motion of translation. The line of action of this force will cut the axis at some point, C, which, in the following, is referred to as the centre of lateral resistance.

The position of this point will generally vary with the angle of inclination, but we do not know according to what law. It has been taken in the following to remain fixed, at a distance, $a$, from the centre of buoyancy, a supposition which does not affect the question of steady motion, but which will necessarily make the solution for disturbed motion deviate to a certain extent from reality.

The force of resistance is resolved into two components, R along the axis, and Q, the lateral resistance, at right angles to the same. Q is taken to vary as sin $\theta$, therefore approximately: $Q = Q_0 \theta$, an assumption which appears justifiable for small values of $\theta$.

The thrust of the propeller, P, acting along the axis.

The pressure of the water on the horizontal rudder, S, supposed acting through the centre of gravity, O, of the rudder area.

The resistance of the rudder, when in its central position, is included in Q. Thus we may put—

$$S = S_0 \alpha_0.$$

The equations of steady motion show that, if the motion is to be stable, we must have—

$$Q_0 a + W \cdot BG > 0.$$

$Q_0 a$, which is denoted by $\beta$, measures the moment of the lateral resistance about the centre of gravity, in the same way as $W \cdot BG$ measures the longitudinal stability. $W \cdot BG$ is in the following denoted by $\epsilon$. It is, in a completely submerged boat, also equal to the transverse stability, and must necessarily always be positive.

$\gamma = \beta + \epsilon$ is a measure of the degree of stability of the motion, and is, in the following, supposed $> 0$.

## Disturbed Motion.

A submarine boat may, while submerged, be subject to various disturbances, which will be here discussed in a general way, and which are in the Appendix treated mathematically.

The principal causes of disturbance are:—Faulty use of horizontal rudder; admission of water through leakages; expulsion of foul air and products of combustion;



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firing of torpedoes and projectiles; movements of crew; existence of free surfaces of liquid; movements of loose weights, such as fuel; variations in buoyancy caused by varying density of sea-water; grounding and collision; variations in speed.

These disturbing elements can hardly all be entirely avoided, and only experience and training can teach the crew of a submarine boat how to counteract them. We may, however, form an idea of their magnitude, and consider the possibility of avoiding or reducing them, and we may, by means of a mathematical investigation, learn the character of the disturbed motion.

We may, thereby, also be enabled to estimate the relative importance of the various sources of disturbance.

Faulty action of horizontal rudder. If, when in steady motion, the rudder is set at an angle $\alpha$, differing slightly from $\alpha_0$, the force of the rudder, S, will vary, both in direction and magnitude. The variation in direction is here disregarded, the angle $(\alpha - \alpha_0)$ being small. The effect of such disturbance may therefore be represented by a vertical force S $(\alpha - \alpha_0)$, acting at a distance $p$ from the centre of gravity.

Admission of water through leakages may, by careful construction of hull and by a strict control over all sea-valves, be reduced to insignificance as long as the boat is not damaged.

Expulsion of foul air need not generally take place during the short period of immersion. Should the necessity arise, however, for drawing on the store of compressed air, and for expelling foul air, the expelled weights will always be small, and the air reservoirs (or chemical stores) used for renewal may be so placed, that no change in the longitudinal balance is caused.

Expulsion of products of combustion, when a motive power is used under water, which, as the various heat-engines, consume both air and fuel. Here the changes in weight would be so serious, that it would become necessary to provide special automatic appliances for always preserving the constancy of the weight and balance.

Electric accumulators are, in this respect, preferable to heat-engines.

Firing of torpedoes from an under-water tube, to which the water has free access before discharge, will cause no sensible change in weight, because the torpedo differs very little in weight from the water, which, after discharge, rushes in and takes its place in the tube, but when a fresh torpedo is afterwards entered into the tube, compensation will be necessary.

Firing of projectiles may cause considerable disturbance, and should be compensated for by the automatic admission of an equal weight of water.

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Movements of crew may be avoided by so arranging the service of the various apparatus and machinery, that everybody has to remain immovable at his post.

Free liquid surfaces may exist in various ways:—

Bilge-water may be avoided by carefully pumping dry before diving. The free liquid surfaces which must exist in tubulous boilers, condensers, and accumulators are insignificant. Free water in ballast tanks is a most serious and likely cause of disturbance, and is discussed separately in a later section. By a free use of air-pipes, draining the air from all corners of the tanks, it must be possible to ensure their being completely filled with water.

Shifting of weights when using a heat-engine for under-water propulsion can hardly be avoided. Feed-water, cooling water, compressed air, liquid fuel, or other weights, would be moved about inside the boat, and would have to be compensated for.

Variation in buoyancy may be caused by navigating in water of varying density, as when passing from a river into the sea, or *vice versâ*. In this way an extra force acting through the centre of buoyancy will be caused, which may amount to 1½ per cent. of the displacement. In cases where such disturbance is likely to occur, a special tank should be provided for compensation.

Grounding. In an ordinary vessel the whole energy of motion is absorbed in the act of grounding, partly in lifting the weight of the ship, partly in friction and damage to the bottom, which actions continue until the vessel is brought to a standstill.

But, in the submarine boat the weight is eliminated by the buoyancy, and the shock is due only to the change of momentum at right angles to the bottom of the sea; the velocity parallel to the bottom remaining almost unchanged, if the angle of incidence is not very great.

Thus the friction, and, therefore, the retardation of speed, will generally be small, and the main result will be a vertical, or nearly vertical force, impulsive in character, which will produce a certain velocity, rotatory and translatory, and will then cease to act.

Collision, which is probably far more dangerous to the submarine boat than grounding, will likewise generally produce an impulsive force.

Variations in speed, besides being caused by grounding and collisions, may also be caused by the act of diving, or by variations in the propelling force. The effect will be a corresponding variation in the resistance and in the action of the rudder, *i.e.*, a variation both in the horizontal and vertical forces which act on the boat.

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It will be seen that nearly all the causes of disturbance here enumerated may be either avoided, or reduced to small magnitude. Consequently, terms involving second, or higher powers of the deviations, or deflections, may be neglected. The assumption that the deflections are small is justified also by the circumstance that, before a deflection has reached any magnitude, it will be counteracted by the rudder.

It is shown that all the disturbances, excepting the three last-mentioned, may be represented in a general way by a vertical force, $f$, and a couple, $f z$, with horizontal transverse axis. In the three last-named cases horizontal disturbances also occur, but their effect will be of interest only in so far as they may indirectly produce changes in the disturbing vertical force and in the disturbing couple.

The disturbed motion will cause variations to take place in the resistance of the water. In steady motion, the inclination of the axis to the trajectory of the centre of gravity is always $\theta_0$. But, in disturbed motion, the boat may turn about a transverse axis, and the centre of gravity may move upwards, or downwards, following a curved and possibly undulating path, whereby a variation in the lateral resistance will be caused.

If the inclination to the horizon at any instant is $\theta$, the angle between the axis and the tangent to the trajectory is now approximately $\theta + \frac{1}{v}\frac{dy}{dt}$, assuming the velocity along the trajectory to be always equal to $v$, an assumption, the justification of which is examined in the Appendix.

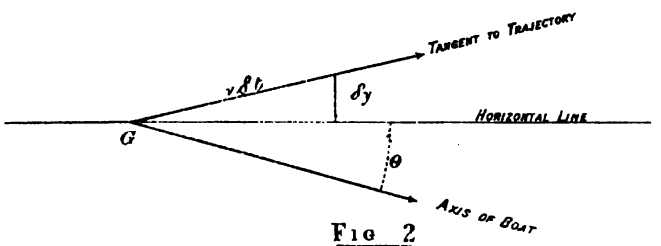

Fig 2

Thus—

$$Q = Q_0 \left(\theta + \frac{1}{v}\frac{dy}{dt}\right)$$

As before, $S = S_0\ a_0$, the rudder remaining untouched in the position it had during steady motion.

Besides the lateral resistance, there will be a resistance depending on angular velocity. The effect of this resistance will be, that the angular motion is extinguished sooner than would otherwise be the case, but it will not influence the general character of the motion. Even if only the resistance varying as $\frac{d\theta}{dt}$ is taken into account, the discussion of the solution will become very much complicated, and it is, therefore, not included in the general investigation, but deferred to a later section.

The simultaneous differential equations of angular and vertical motion contain, according to the foregoing, second differential co-efficients of $\theta$ and $y$ (the vertical ordinate), first differential co-efficient of $y$ and the first power of $\theta$.

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The general solution of these equations gives expressions for $y$ and $\theta$, which, besides a constant term, each contain three exponential terms, and the expression for $y$ contains, moreover, a term with $t$ as factor.

The exponents are given by an equation of the third degree, which really determines the character of the motion. It is found that if $\varepsilon$ and $\beta$ are both positive, all the exponential terms will vanish, and the motion will be stable. If either $\varepsilon$ or $\beta$ are negative, *i.e.*, if the centre of gravity falls above the centre of buoyancy, or, if the centre of lateral resistance falls forward of the centre of gravity, the motion will be unstable ; for then there will always be one real positive exponent, showing an ever increasing motion, or there will be two imaginary exponents with positive real parts, showing ever increasing oscillations.

In the general investigation, $\varepsilon$ and $\beta$ are supposed both positive, *i.e.*, stable motion.

The cases $\varepsilon = 0$ and $\beta = 0$ are treated separately.

## General Character of Disturbed Motion.

Suppose a submarine boat moving with steady motion along a horizontal line, while inclined at an angle $\theta_0$ to the horizon.

If now a disturbance occurs, consisting of a permanent vertical force acting at any point of the axis of the boat, and if the rudder is not touched, she will begin to turn towards a new position of angular equilibrium :

$$\theta_1 = \theta_0 + \frac{f(a + z)}{\varepsilon}.$$

This position may be reached either through a series of oscillations, decreasing in amplitude, or through a steady swing.

The limiting deflection is directly proportional to the moment of the disturbing force about the centre of lateral resistance, and inversely proportional to the longitudinal stability.

The centre of gravity will at the same time follow a path, which is at first curved upwards or downwards, possibly undulating, and ending in a straight line inclined to the horizon, showing a limiting vertical velocity :

$$\left(\frac{dy}{dt}\right)_1 = \frac{f}{Q_0} v + \frac{f(a + z)}{\varepsilon} v.$$

The first of these terms is due to the direct action of the disturbing force taken to act

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at the centre of lateral resistance; the second term is due to the deflection, it may easily be many times greater than the first term.

It is seen that the vertical limiting velocity is directly proportional to the disturbing force, and to the speed, and that it increases with a decrease in lateral resistance and in longitudinal stability.

Moreover, forces acting forward of the centre of lateral resistance will cause a greater vertical motion than forces acting aft of that point, for the two terms in the expression for $\left(\frac{dy}{dt}\right)_1$ will in the former case be of the same sign, in the latter case of opposite signs.

It may be concluded that it is important to have a good longitudinal stability, and to avoid any change in the longitudinal balance. Movements of weights from aft to forward, and addition of weights forward, are in particular dangerous, because they will cause the boat to descend. Curves 1 and 2 (see Plate XLVII.) show the path of the centre of gravity when a disturbing downward force is acting forward and aft respectively.

Whether, in the act of passing from the equilibrium of steady motion to the limiting condition, the motion be oscillatory or not, is shown in the Appendix to depend in the first instance on the relation between a certain factor F and the longitudinal stability, $\varepsilon$ —

$$F = \frac{Q_0^2 \rho^2}{M v^2},$$

where $\rho$ is radius of gyration and M is the mass of the boat. F depends, therefore, on the shape, size, and speed, and, as $Q_0$ varies probably nearly as $v^2$, F will be large in long flat boats with high speed.

The condition that non-oscillatory motion shall be at all possible is found to be—

$$F > 27\,\varepsilon.$$

Let us suppose this to be the case, and let us imagine C, the centre of lateral resistance, to move from forward to aft in such a way that only $a$ is varied, but all other elements of the boat, and therefore also F and $\varepsilon$, remain unaltered.

When C is forward of the centre of gravity, G, the motion will be oscillatory and increasing; *i.e.*, the motion is entirely unstable. When C coincides with G, the oscillations will neither increase nor decrease. When C lies aft of G, the oscillations will be decreasing.

As C moves farther aft, the moment of lateral resistance $\beta$ increases, and therefore

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$\gamma = \beta + \varepsilon$ increases. When $\gamma$ has reached a certain value, the motion will cease to be oscillatory, and the boat will turn steadily from one position of equilibrium to the other.

Moving C further aft, another point will be reached, where the motion becomes again oscillatory, and will continue to be so, however much $\gamma$ is increased.

Thus there would be three phases in the character of the motion, if C were imagined to be moved from forward to aft.

During the first phase the boat will oscillate, and the oscillations, which will be of great amplitude and long period, are mainly governed by the longitudinal stability, $\varepsilon$. During the second phase the boat will swing steadily to the new position without oscillations. During the third phase the boat will again oscillate, but the oscillations are now governed mainly by the moment of the lateral resistance, and the position of equilibrium will be approached in a series of small oscillations of short period.

For the steering of the submarine boat, it would appear to be desirable to approach or to attain the non-oscillatory condition. We must therefore secure, firstly, the inequality $F > 27\,\varepsilon$, and, secondly, a sufficiently high value of $\gamma$. We shall examine how this may be done in various cases.

It is likely that, in practice, the motion will nearly always be found non-oscillatory, even in such cases where oscillations might be expected to take place. The reason for this is, probably, that the rudder is generally used before the return swing has commenced, and that the angular resistance extinguishes the motion.

## High Speed Boats.

For boats, such as are now generally constructed, of from 100 to 200 tons displacement, and with our present means of under-water propulsion, a speed of above 12 knots must be considered "high speed."

In such boats F will generally be found much greater than $27\,\varepsilon$, not only on account of the high value of $v$, and therefore of $Q_0$, but also because such boats must be of great length, and therefore of great radius of gyration.

We can therefore afford to make $\varepsilon$ large, which is the more desirable in a high-speed boat, because the vertical movements vary mainly as $\frac{v}{\varepsilon}$. But great longitudinal stability means great depth, and as the lateral resistance, Q, is already large in virtue of the speed, we need not make the breadth great.

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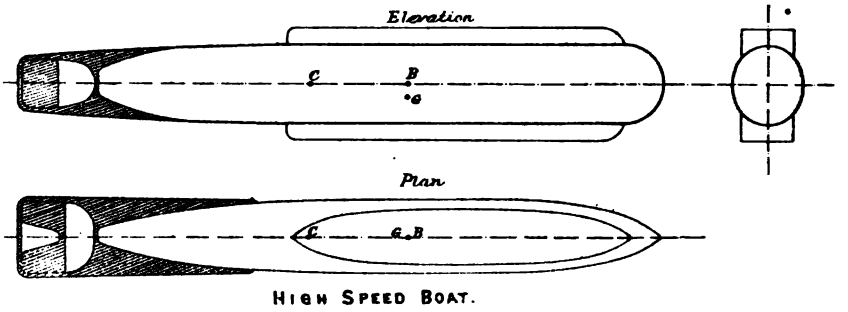
Thus we arrive at a long, deep, and comparatively narrow boat, with great metacentric height.



HIGH SPEED BOAT.
FIG. 3.

Moreover $\gamma$ must have a certain high value, and therefore $a$ must be made very great by drawing the centre of gravity forward and the centre of lateral resistance aft. This leads to a deep narrow fore-body, a flat and broad aft-body, and may be attained by a shape, such as is indicated diagrammatically on Fig. 3, fitting large horizontal fins aft.

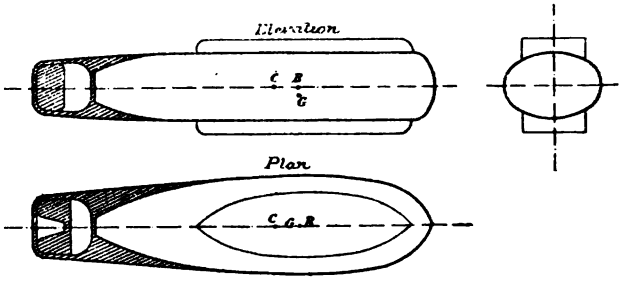
The French submarine boats, *Gustave Zédé*, *Morse*, and others approach to this type. (*Morse* : $\frac{\text{Length}}{\text{Breadth}} = 13$; speed 12·3 knots).

## LOW SPEED BOATS.

These may be defined as boats of less than 6 knots.

The longitudinal stability need not be so great in low-speed as in high-speed boats. The inequality, $F > 27\,\epsilon$, may, therefore, be attained by somewhat reducing $\epsilon$. Moreover, F may be increased by making the boat broad and flat.

Great length is not necessary in a low-speed boat, and is, on the whole, objectionable in point of internal arrangements and weight of hull. It will be found that a reduction in $\epsilon$ will cause a reduction in the value of $\gamma$, for which non-oscillatory motion is reached.



LOW SPEED BOAT.
FIG. 4.

Thus, it is not required in a low-speed boat to draw the centre of lateral resistance so far aft of the centre of gravity as in the high-speed boat. The low-speed boat should, therefore, be of short length, small depth, and great breadth. The shape being given, the metacentric height should be made as great as possible by a low centre of gravity. Large horizontal fins should be placed aft. The general character of the type is indicated on Fig. 4.

The "Holland Torpedo Boat Company's" first boat, the *Holland*, approaches the

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low-speed type here described $\left(\frac{\text{Length}}{\text{Breadth}} = 5{\cdot}2\text{; speed, 6 to 7 knots}\right)$, but differs in having greater depth and stability.

For doing service off the coasts in deep water, and for semi-offensive purposes, the high-speed type appears best suited. For waters of limited depth and extension, such as estuaries, straits, and sounds, a type between the two extremes would be well adapted. For use in roadsteads and ports, where the depth of water is very limited, where mine-fields make small draught desirable, and where the condition of the sea bottom permits sliding along the bottom when found expedient, the flat, low-speed would appear to be the best.

### Effect of Free Liquid Surfaces.

The most likely case of free liquid surfaces occurring, and one which has probably caused the failure of many submarine boats, is that of free water in ballast tanks. The effect is practically the same as if the stability had been diminished. It may be noticed that, in this way, the longitudinal stability may be seriously reduced, while the transverse stability remains practically unaltered; if, namely, the tanks are of small breadth, but of great length.

As the longitudinal stability is of utmost importance for the good behaviour of a submarine boat, the great danger of free liquid surfaces is obvious.

The main ballast tank should, therefore, be subdivided into smaller compartments, and each compartment should, when submerged, be kept either completely filled or completely empty. Two deep end tanks may be used for regulating the trim, and for providing for smaller fluctuations in displacement. The water in these tanks must, unavoidably, have free surfaces, but the length of the tanks may be reduced to a minimum.

### Centre of Lateral Resistance Coincides with Centre of Gravity.

Here $\beta = 0$. The angular motion will be stable and purely oscillatory. Were it not for the angular resistance, the oscillations would, in this case, never be extinguished.

### Longitudinal Stability Vanished.

Here $\epsilon = 0$ and $\gamma = \beta$. As mentioned, this case may occur when there is free water in ballast tanks, and is of interest also because the equations apply, with few modifications, to motion in a horizontal plane where $\epsilon$ is likewise equal to zero. The



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investigation shows that the motion is always non-oscillatory as long as $\beta < \frac{F}{4}$; but, if the centre of lateral resistance falls more aft, than given by this inequality, *i.e.*, if $a < \frac{Q_0 \rho^2}{4 M v^2}$, the motion will be oscillatory.

The equation for $\theta$ shows that $\theta$ will be ever increasing with a constant angular velocity if a permanent disturbing force is acting. The equation for $y$, the vertical ordinate, shows an accelerating vertical velocity. The equations, consequently, hold only during the first stages of the motion, while $\theta$ is still small.

The deviations due to a given disturbance are smaller the greater $a$ is, and the greater the speed.

The motion is stable as long as $\beta$ is positive. If besides $\epsilon = 0$, also $\beta = 0$, the boat will be in a state of neutral equilibrium; $\theta$ will vary directly as $t^2$, and the expression for $y$ will contain terms with $t$, $t^2$, and $t^3$.

For motion in the horizontal plane, *i.e.*, for steering of ordinary vessels, the equations hold almost unaltered, as regards small involuntary deviations from the course, and during the first part of the turning movement, when the rudder is used. $\beta$ is here referred to the vertical plane.

That stability of motion may exist in case of horizontal steering is evidenced by the Thornycroft torpedo-boats with double spade rudders. These boats may, in calm weather, when going at high speed, keep their course almost unaltered for a quarter of an hour or more, without the rudders being touched. It is probable that the effect of the propeller race on the rudders is such as to make $\beta$ positive in this case.

If $\beta$ is negative, as is probably most often the case in the vertical plane in ordinary ships, the motion is unstable, and the rudder must, therefore, generally be used continually in order to keep the ship on her course. The lateral deviations which are hereby incurred are, however, in most cases of small consequence. It is generally immaterial whether the ship moves along one line, or along another parallel to the original at several fathoms' distance.

In narrow passages, where such deviations are not permissible, great care must be exercised in order to keep the ship on the desired track.

A submarine boat is in the vertical sense always, so to speak, manœuvring through a narrow passage; a deviation of a few fathoms upwards may bring it to the surface, where it may be exposed to the attack of the enemy; a deviation downwards may carry it to the bottom, or expose it to crushing pressures.

The case of an ordinary ship turning in a circle with steady velocity is discussed in

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the Appendix, on account of its analogy to the limiting disturbed motion of a submarine boat.

The equations for initial motion show that when the rudder is placed aft, the centre of gravity will always begin to move out of the course line to the opposite side of that to which the ship is ultimately turning. This holds both in case of vertical and horizontal steering, and is a fact which has often been observed, when laying down the turning circles of ordinary vessels.

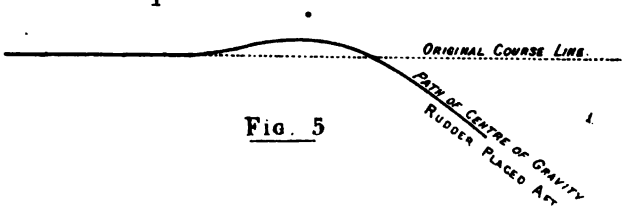

Fig. 5

## Impulsive Forces.

By grounding and collision a submarine boat may be subject to forces of impulsive character. Apart from the damage which the boat may suffer, the effect will be that the boat will shift to a different level, where it will proceed in a horizontal line, inclined at the original angle $\theta_0$ to the horizon. Before the boat comes to rest in its new position of equilibrium, it may perform some oscillations, or it may move in a steady double swing.

The case of pure unresisted oscillations, imagined to be produced by impulsive action, is treated in the Appendix; it is of interest for comparison with the more complex case of resisted motion.

## Angular Resistance Included.

This resistance must be in character similar to the resistance to rolling of an ordinary vessel, the main point of difference being that the pitching movements of the submarine boat are of much smaller amplitude than the rolling of ordinary vessels.

The late Mr. W. Froude has shown, that at small angles the greater part of the resistance to rolling varies as the first power of $\frac{d\theta}{dt}$, the smaller part as the second power. It appears, therefore, justifiable to neglect the resistance varying as $\left(\frac{d\theta}{dt}\right)^2$, and to put the moment of resistance equal to $k\frac{d\theta}{dt}$.

It is found that the form of the expressions for $\theta$ and $y$ remains unaltered, and only the co-efficients and the exponents of the vanishing terms change value. The limiting inclination and vertical velocity will, however, be the same as if the resistance $k\frac{d\theta}{dt}$

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had not existed, but the limiting condition will be sooner reached, and the oscillatory motion will probably in most cases not be perceptible.

It is found, on account of this resistance, that we may have stable motion even if the centre of lateral resistance is situated, up to a certain distance, forward of the centre of gravity; this distance depending on the magnitude of the angular resistance.

### Stability of Motion.

This quality may be defined as the tendency of the boat to preserve the state of steady motion.

It has already been mentioned that, although $\gamma > 0$ is a necessary condition for stability of motion, it is not sufficient. If, namely, one of its component parts, $\beta$ and $\epsilon$, is negative, the motion will be unstable.

By imagining the boat to be deflected a small angle from its position of steady motion, and then left to itself, it is proved in the Appendix that, even if one of the quantities, $\beta$ and $\epsilon$, is equal to zero, the other remaining positive, the motion will still be theoretically stable.

Thus the general conditions of stability of motion, disregarding angular resistance, are—

$$\beta \geqq 0, \quad \epsilon \geqq 0 \quad \text{and} \quad \gamma > 0;$$

*i.e.*, the centre of lateral resistance must not fall forward of the centre of gravity, and the centre of gravity not above the centre of buoyancy. If either of these conditions is not fulfilled, the motion will be unstable.

Although, as in horizontal steering, it may be possible to keep the depth by means of the rudder, even if $\beta$ is negative, it appears safest, at any rate in experimental boats, to draw the centre of lateral resistance as far aft as possible, and to place the centre of gravity well forward.

This leads to a narrow, deep fore-body, broad and flat aft-body, and large horizontal fins, with rudders placed as far aft as possible. High speed is favourable to stability of motion, but should always be accompanied by great longitudinal stability.

### Manœuvring Power.

Manœuvring power may be defined as the power to acquire and to extinguish angular velocity. This quality must be essentially different from the same quality in the horizontal plane, from the existence of the longitudinal stability.

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When the vertical rudder is used, the boat acquires a certain angular velocity, and, as long as the rudder acts, it will go on turning indefinitely. But, when the horizontal rudder is used, the movement will, apart from resistances, be limited by the longitudinal stability.

The vertical turning movements which have to be carried out in a submarine boat, are small compared to the horizontal movements of any vessel. Although it is desirable that these movements should be effected quickly, it will, on account of their smallness, not be necessary to use great angles of inclinations, which, moreover, for other reasons will be inconvenient.

Inclinations exceeding 10° will rarely be used. It is clear, therefore, that great angular velocities cannot, and need not, be attained during the manœuvres. The manœuvring power should be secured exclusively by increasing the rudder area, and by placing it far aft; hereby the stability of motion is at the same time somewhat increased. The faculty of steering steadily in a given direction, and of readily extinguishing turning movements, appears more important in a submarine boat than the faculty of quick and easy turning. Stability of motion should, therefore, not be sacrificed in order to obtain manœuvring power. The opposing influence of resistance will be small, and will be rather beneficial, because it will hardly be felt before the rudder is to be eased, and will then assist in bringing the boat to rest in the new direction.

A small moment of inertia is favourable to manœuvring power, but this element must be settled principally with regard to internal arrangements, speed, &c.

## Surplus Buoyancy.

It is desirable to retain a certain amount of surplus buoyancy in a submarine boat. It will provide a margin of safety against minor leakages. If the propelling machinery breaks down, the boat will at once ascend to the surface. When on the surface the boat will possess a certain amount of freeboard, without need of pumping. Generally the surplus buoyancy will reside in the conning tower. The surplus buoyancy is not, as some people imagine, kept for the sake of stability; its influence on stability will generally be small, and may even be detrimental: if, namely, it is produced by removing a weight from below the centre of gravity.

The facility with which the boat dives from "awash," must be dependent on the smallness of the angle $\theta_0$, which it has to maintain when in steady motion under water. It is evident that both as regards resistance and convenience this angle should be as small as possible for a given surplus buoyancy, *i.e.*, $\frac{\theta_0}{w}$ should be small.

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The investigation shows that this ratio is smallest when the conning tower is placed on the same side of the centre of gravity as the rudder.

Moreover, that the best result is obtained when the rudder is placed forward; and finally, that great central horizontal area and high speed are favourable to a great surplus buoyancy.

### The Rudders.

As in any other ship, a powerful steering gear, great area and leverage of rudder, and high speed are elements conducive to good steering qualities in the submarine boat.

As regards the position of the rudders opinions are divided. Some hold that they should be placed forward, others that they should be placed aft, others again both forward and aft, while some inventors combine the fore or aft rudders with midship rudders.

It has already been stated that it is favourable in point of surplus buoyancy and inclination to place the rudders forward. As also forces acting forward produce greater vertical motion than when acting aft, we may conclude that forward rudders must be more effective than aft rudders.

When plunging from the condition "awash," fore rudders are not liable to get out of the water as aft rudders are, and the direct action of the fore rudders will assist the vertical movement, while the upward pressure on aft rudders will counteract the same. It appears, indeed, that with some submarine boats, notably long ones, difficulties have been experienced in effecting the dive using aft rudders only.

In spite of all these arguments in favour of placing the rudders forward, this disposition can hardly be recommended except in very long boats, where it may prove a necessity. The same reasons which, in an ordinary ship, make us place the rudder aft, namely, that it is there better protected against damage and fouling, and that its effect is augmented by the action of the propeller race, hold good in a still higher degree in a submarine boat.

The drawbacks which are connected with placing the rudders aft, may be almost entirely obviated by giving them great area and leverage, and by placing the conning tower aft of the centre of gravity.

It has already been mentioned that stability and steadiness of motion are increased by aft rudders.

Midship rudders produce no rotation, and must, therefore, be combined with either fore or aft rudders. They counteract the surplus buoyancy, and facilitate diving, but their position is very exposed, and they complicate the service of the boat. It

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appears preferable, therefore, to place rudders aft in all cases, and in very long boats, where experiments might show the necessity thereof, also to place rudders forward; but in such case the forward rudders should be used for steering, the aft rudders only as inclined movable planes. The combination of four rudders, two forward, and two aft, is used in the French submarine boat *Le Narval*, and probably in several, if not all, other French boats. The American boat, the *Holland*, has only rudders aft.

It has already been mentioned, that steering in the vertical plane is more difficult than in the horizontal plane; the helmsman, who in the latter case is concerned only with the angular motion, is in the submarine boat concerned both with the angular and the vertical motion. Moreover, if a disturbance of permanent nature occurs in a submarine boat, the helmsman has to find the new position of equilibrium both for the boat and for the rudder.

It is obvious that only great experience and practice, and the use of good instruments, can enable the helmsman to solve his task satisfactorily. In an experimental boat a long and careful training of the helmsman, who is to work the horizontal rudders, must, therefore, be an essential condition of success, unless automatic appliances are devised which make his presence superfluous.

### Effect of Non-Symmetry.

The general conclusion of the discussion on this point is that symmetry should, as far as possible, be preserved, at any rate in an experimental boat, because the effect of non-symmetry will vary with the speed and immersion, and is, therefore, likely to render steering difficult.

If symmetry is not preserved, whether it be in the shape of hull or position of propeller shaft, the vertical component of the resulting extra force should always act downwards, and the force should have the greatest possible turning moment about the stock of the horizontal rudder, opposite, in sign, to that of the surplus buoyancy; hereby the ratio $\frac{\theta_0}{w}$ will be the smallest possible.

### Effect of Heeling.

When a submarine boat is inclined transversely, the vertical rudder will act partly as a horizontal one and conversely. Apart from distribution of weights, heeling may be caused by the action of the propeller; this is, in the Whitehead torpedo, avoided by using two screws turning in opposite directions. In a submarine boat, the turning moment of the propeller will, with the present means of under-water propulsion, probably be insignificant.

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### On Even Keel.

The principle of keeping the boat always on even keel, not only during steady motion, but also when rising or sinking, has by many inventors, and formerly, when the Whitehead mechanism was yet unknown, also by the author of this paper, been recommended as the safest. The depth may then be kept either by means of inclined planes or fins—probably used in some of the French boats (*Le Narval*)—or by downhaul propellers, or by pumping water in and out of the boat, all of which methods are discussed in the Appendix.

Since the Whitehead mechanism has been made public, there appears to be no necessity to use any of these methods when under way, for what has been done in the Whitehead torpedo, we must also be able to do in a submarine boat, where the pendulum-valve mechanism will act under more favourable conditions than in the torpedo, not being exposed to the violent accelerations of the latter.

We now know that steering has been effected successfully by the inclining method, using horizontal rudders only, namely, in the *Holland*.

The Whitehead mechanism may be used either as an indicator, or for automatically working the rudders through a servo-motor.

For keeping a certain depth at rest under water, a pump may be used. This has been done successfully in *Le Goubet*. When under way, this system does not work well, at any rate in large boats.

The mathematical investigation in the Appendix holds, in these cases, with the modification, that $\theta_0$ is put equal to zero.

### Steering by Shifting Weights inside the Boat.

The shifting of weights may consist in pumping water from one end tank to the other. By this method all danger of damage and fouling of rudders is avoided, and the system appears simple and effective. It is shown in the Appendix that pure couples are very effective in producing vertical motion.

It is doubtful, however, whether sufficiently great turning moments can be created in this way without using a very powerful pump, involving comparatively great expenditure of power.

### Regulating and Balancing.

Before diving, the displacement and the trim should be so regulated that, when at rest the boat should lie upright on an even keel, with only that part of the conning tower which corresponds to the surplus buoyancy, projecting above the water. Let

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this be called "the diving condition." Experience may show that it is advantageous to trim the boat somewhat on the keel in this condition.

Moreover, every precaution should be taken to preserve the displacement and the position of the centre of gravity unaltered during submergence.

The former of these operations, the "balancing," is performed while in the light condition by means of the ballast tanks. It has been stated that there must be at least one large (possibly subdivided) "bottom tank," and two smaller "end tanks."

Let us suppose that, by experiment, the draught and trim have been determined, from which the boat may be brought directly to the diving condition, by simply filling the bottom tank. Let this be called the "balanced trim," and the condition, the "normal light condition."

By placing the bottom tank well forward, the balanced trim will be such that the boat is well on the keel, the screw well immersed, and the fore-body well out of the water. It is, therefore, a water-line well adapted for navigation.

Suppose now the boat to be approaching the enemy. It should then be kept nearly on its balanced trim until in danger of being discovered. By means of the end tanks the boat should be brought exactly on that trim, whereafter the bottom tank, which has hitherto been kept completely empty, is completely filled. The boat will now be in her "diving condition," ready to dive at any moment.

Before going to dive, it should be ascertained, that the bottom tank is completely filled, that all bilge-water is pumped out, and all loose weights fastened. The boat should be laid exactly upright. Each man should be assigned a certain position, from which he should not move during the dive. The balancing process here described is, in principle, the same operation which must be carried out with every Whitehead torpedo, and for the same reasons.

If, during the dive, a permanent disturbance occurs, it will, within certain limits, be possible to meet it by the action of the rudder alone; the result being that steady motion is again established, but the angle of rudder and the angle of inclination are generally both altered. If the change in rudder angle and inclination is observed to be considerable, it appears advisable either to stop and rebalance the boat, or to keep going and attempt to bring the boat back to the original inclination by pumping water from one end tank to the other.

If after this is done the boat shows a tendency to sink or to rise, it is a sign that a weight has been added or removed, and water should accordingly be pumped either out of or into the boat.

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At the present stage of submarine navigation submerged runs of more than a couple of miles will, probably, rarely be undertaken. For such short distances it would appear possible to avoid permanent disturbances of such magnitude, that they could not be dealt with by the rudder alone. The method here described of balancing under way may, therefore, not find any use for the present.

If the sea is too rough for the exact determination of the balanced trim, the balancing operation may, possibly, be performed under water when at rest, by first making the boat float in equilibrium on even keel, and then pumping out a certain amount of water in order to produce the surplus buoyancy.

## Summary of Conclusions.

Shape.—The fore-body should be deep, the aft-body broad and flat. Large horizontal fins placed aft. High-speed boats should be longer, deeper, and of smaller breadth than low-speed boats. Symmetry about a horizontal central plane should, as far as possible, be preserved. If non-symmetry is unavoidable, the extra force thereby created should have a vertical component, acting downwards, and the force should have the greatest possible turning moment about the stock of the horizontal rudder.

Stability should, the type being given, be made as great as possible, by securing a low centre of gravity.

The horizontal rudder should be large, and placed as far aft as possible.

Surplus buoyancy should be as great as found convenient. The conning tower should be placed aft of the centre of gravity.

The displacement, and the longitudinal distribution of weights, should be preserved strictly unaltered while submerged, and should be carefully regulated before diving. Unavoidable disturbances should occur aft rather than forward.

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I regret that I have been unable to base this paper more on practical experience and less on theory than has been the case. So far, I have not myself had an opportunity of constructing or manœuvring submarine boats, and I have been obliged, therefore, to go entirely by theoretical considerations, and by accounts which have been published regarding past and present submarine boats.

I wish, therefore, to apologise beforehand, if points are found in this paper on which I have laid particular stress, and which may hereafter in practice be proved to be of small importance; and, likewise, if I have treated lightly, or even omitted, points which later experience may show to be of greater importance.

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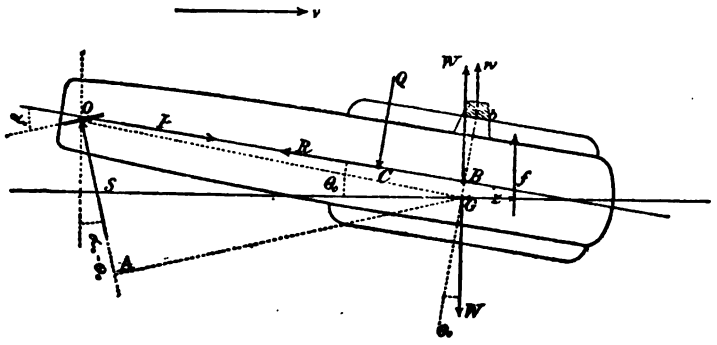
## APPENDIX.

### Steady Motion.

Referring to the figure: B G must be a small quantity compared to G O, and O will lie practically in the axis, except for very great rudder angles. Drawing G A at right angles to the line of action of S, we may, therefore, approximately, put angle O G A = $a$. Let G O = $p$.



In forming the equations of motion, we refer here, and in the following, to a system of rectangular co-ordinates, fixed in space, with origin in the initial position of the centre of gravity of the boat.

The axis of X is horizontal and positive to the right. The axis of Y, vertical and positive upwards. The angles are measured from the positive direction of the axis of X, positive in direction of the hands of a watch.

Resolving, horizontally—

$$M \frac{d^2 x}{d t^2} = P - R - S_0 a_0 (a_0 - \theta_0) - Q_0 \theta_0^2 = 0.$$

Neglecting squares of the small angles, we find—

$$P = R. \tag{1}$$

Resolving, vertically—

$$M \frac{d^2 y}{d t^2} = S_0 a_0 + w - Q_0 \theta_0 = 0. \tag{2}$$

Taking moments about G—

$$M \rho^2 \frac{d^2 \theta}{d t^2} = S_0 p a_0 - Q_0 \theta_0 a - W . BG . \theta_0 + (P - R) . BG - w . Gb . \theta_0 = 0.$$

As $w$ is a small quantity, we may neglect the last term, and, substituting, we get—

$$S_0 p a_0 = \theta_0 (\beta + \epsilon) = \theta_0 \gamma. \tag{3}$$

From (2) and (3) we find—

$$\theta_0 = \frac{w p}{\delta} \tag{4}$$

and—

$$a_0 = \frac{w \gamma}{S_0 \delta} \tag{5}$$

where—

$$\delta = Q_0 (p - a) - W \times BG.$$

The turning moment being $S_0 p a_0 - \theta \gamma$, we see that any increase in $\theta$ will cause the turning moment to decrease as long as $\gamma$ is positive, but to increase if $\gamma$ is negative. Conversely for a decrease in $\theta$. $\gamma > 0$ is, therefore, a necessary condition of stability of motion.

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## Disturbed Motion.

### *Formation and Solution of the Equations of Motion.*

The boat is supposed to be in steady motion, when at time $t = o$, a small vertical force $f$ begins to act upwards (positive), at a distance $z$ forward of G, thus producing a negative turning moment. The rudder is not touched, and remains at the angle $a_0$.

Resolving, horizontally—

$$M \frac{d^2 x}{d t^2} = P - R - S_0 a_0 (a_0 - \theta) - Q_0 \left(\theta + \frac{1}{v} \frac{d y}{d t}\right) \theta.$$

Considering only small disturbances, we may assume the horizontal acceleration to be zero, and, neglecting small terms of second order, we get—

$$P = R. \tag{6}$$

Resolving, vertically—

$$M \frac{d^2 y}{d t^2} = S_0 a_0 + w + f - Q_0 \left(\theta + \frac{1}{v} \frac{d y}{d t}\right).$$

From steady motion—

$$S_0 a_0 + w - Q_0 \theta_0 = 0.$$

Subtracting—

$$\frac{d^2 y}{d t^2} + \frac{Q_0}{M v} \frac{d y}{d t} + \frac{Q_0}{M} \theta = \frac{Q_0 \theta_0 + f}{M}. \tag{7}$$

Taking moments about G—

$$M \rho^2 \frac{d^2 \theta}{d t^2} = S_0 p a_0 - \beta \left(\theta + \frac{1}{v} \frac{d y}{d t}\right) - \theta + (P - R) BG - w . G b \theta - f z.$$

Neglecting small terms, and substituting—

$$\frac{d^2 \theta}{d t^2} + \theta \frac{\gamma}{M \rho^2} + \frac{\beta}{M \rho^2 v} \frac{d y}{d t} = \frac{\gamma \theta_0 - f z}{M \rho^2}. \tag{8}$$

The general solution of the two simultaneous differential equations (7) and (8) is found to be—

$$\theta = A_0 + A_1 e^{\lambda_1 t} + A_2 e^{\lambda_2 t} + A_3 e^{\lambda_3 t} \tag{9}$$

$$y = B_0 + B_1 e^{\lambda_1 t} + B_2 e^{\lambda_2 t} + B_3 e^{\lambda_3 t} + \left[\frac{f}{Q_0} + \frac{f(a + z)}{\varepsilon}\right] v t. \tag{10}$$

Where the A's and B's are constants, determined by the initial conditions, and by the mutual relations which exist between them, and where $\lambda_1$, $\lambda_2$, $\lambda_3$ are roots in the equation—

$$\lambda^3 + \lambda^2 \frac{Q_0}{M v} + \lambda \frac{\gamma}{M \rho^2} + \frac{\varepsilon Q_0}{M^2 \rho^2 v} = 0. \tag{11}$$

### *Determination of Constants.*

By substitution in (8) from (9) and (10), we find—

$$A_0 = \theta_0 - f \frac{a + z}{\varepsilon} \tag{12}$$

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$$B_1 = -A_1 \frac{\lambda_1^2 M \rho^2 v + \gamma v}{\beta \lambda_1} \tag{13}$$

$$B_2 = -A_2 \frac{\lambda_2^2 M \rho^2 v + \gamma v}{\beta \lambda_2} \tag{14}$$

$$B_3 = -A_3 \frac{\lambda_3^2 M \rho^2 v + \gamma v}{\beta \lambda_3}. \tag{15}$$

The initial conditions are :—

$$t = 0, \quad \theta = \theta_0, \quad \frac{d\theta}{dt} = 0, \quad y = 0, \quad \frac{dy}{dt} = 0,$$

Substituting these in (9) and (10), we find the following relations between the constants—

$$A_1 + A_2 + A_3 = f\frac{a + z}{\epsilon} \tag{16}$$

$$A_1 \lambda_1 + A_2 \lambda_2 + A_3 \lambda_3 = 0 \tag{17}$$

$$A_1 \lambda_1^2 + A_2 \lambda_2^2 + A_3 \lambda_3^2 = -\frac{f z}{M \rho^2} \tag{18}$$

$$\frac{A_1}{\lambda_1} + \frac{A_2}{\lambda_2} + \frac{A_3}{\lambda_3} = \frac{\beta}{\gamma v} B_0. \tag{19}$$

We have thus four equations for the determination of A , $A_2$, $A_3$, and $B_0$.

Put—

$$\frac{a + z}{\epsilon} = K$$

and—

$$-\frac{f z}{M \rho^2} = L,$$

we then find, by solving the equations—

$$A_1 = \frac{L + K \lambda_2 \lambda_3}{(\lambda_1 - \lambda_2)(\lambda_1 - \lambda_3)} \tag{20}$$

$$A_2 = \frac{L + K \lambda_1 \lambda_3}{(\lambda_2 - \lambda_3)(\lambda_2 - \lambda_1)} \tag{21}$$

$$A_3 = \frac{L + K \lambda_1 \lambda_2}{(\lambda_3 - \lambda_1)(\lambda_3 - \lambda_2)} \tag{22}$$

and—

$$B_0 = \frac{\gamma v}{\beta}\left(\frac{A_1}{\lambda_1} + \frac{A_2}{\lambda_2} + \frac{A_3}{\lambda_3}\right). \tag{23}$$

Thus all the constants may be determined when the λ exponents are known.

It will be noticed that the disturbing force occurs as a factor in all the constants, while the exponents are independent of this force.

*Determination of Exponents.*

Returning to the equation (11)—

$$\lambda^3 + \lambda^2 \frac{Q_0}{M v} + \lambda \frac{\gamma}{M \rho^2} + \frac{\epsilon Q_0}{M^2 \rho^2 v} = 0.$$

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Put—

$$\lambda = x - \frac{Q_0}{3\,M\,v}$$

and the equation reduces to—

$$x^3 + x\,A + B = 0$$

where—

$$A = \frac{\gamma}{M\,\rho^2} - \frac{Q_0^2}{3\,M^2\,v^2}$$

and—

$$B = \frac{2}{27}\,\frac{Q_0^3}{M^3\,v^3} - \frac{Q_0\,\gamma}{3\,M^2\,\rho^2\,v} + \frac{\varepsilon\,Q_0}{M^2\,\rho^2\,v}.$$

The roots of this equation are—

$$x_1 = p_1 + q_1$$
$$x_2 = p_1\,a_1 + q_1\,a^2$$
$$x_3 = p_1\,a^2 + q_1\,a$$

where—

$$a = -\frac{1}{2} + \frac{\sqrt{-3}}{2}$$

$$a^2 = -\frac{1}{2} - \frac{\sqrt{-3}}{2}$$

and $p_1$ and $q_1$ are determined by—

$$p_1^3 = -\frac{B}{2} + \sqrt{\frac{B^2}{4} + \frac{A^3}{27}}$$

$$q_1^3 = -\frac{B}{2} - \sqrt{\frac{B^2}{4} + \frac{A^3}{27}}$$

and, therefore—

$$\lambda_1 = p_1 + q_1 - \frac{Q_0}{3\,M\,v} \tag{24}$$

$$\lambda_2 = p_1\,a + q_1\,a^2 - \frac{Q_0}{3\,M\,v} \tag{25}$$

$$\lambda_3 = p_1\,a^2 + q_1\,a - \frac{Q_0}{3\,M\,v} \tag{26}$$

from which the three exponents may be determined.

As regards the character of these exponents, we see directly, from (11), that as long as $\gamma$ is positive, which has been shown to be necessary for stable motion, all real roots must be negative. We know, moreover, that in an equation of the third degree there will be either one real and two imaginary roots, namely, when the discriminant, $\frac{B^2}{4} + \frac{A^3}{27}$ is positive; or three real roots, of which two are equal, namely, if the discriminant is zero, and three real and unequal roots, if it is negative.

*Examination of the Discriminant.*

$$\frac{B^2}{4} + \frac{A^3}{27} = \frac{1}{27\,M^3\,\rho^6}\left[\gamma^3 - \frac{F}{4}\gamma^2 - \frac{9\,F\,\varepsilon}{2}\gamma + \frac{27\,F\,\varepsilon^2}{4} + F_\varepsilon^2\right]$$

putting—

$$\frac{Q_0^2\,\rho^2}{M\,v^2} = F.$$

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The discriminant has the same sign as the expression inside the brackets, which may be denoted D. Thus—

$$D = \gamma^3 - \frac{F}{4}\gamma^2 - \frac{9\,F\epsilon}{2}\gamma + \frac{27\,F\epsilon^2}{4} + F^2\epsilon \qquad (27)$$

F depends on shape, size, and distribution of weights, and on the speed, and is always positive. If we take $Q_0$ to vary as $v^2$, F will vary as $v^2$. Both $\epsilon$ and $\gamma$ are likewise positive.

We shall now examine how D varies with variation of $\gamma$, keeping $\epsilon$ unaltered, *i.e.*, by imagining the centre of lateral resistance to move.

When $\gamma$ is negative, D will begin at $-\infty$; and, as $\gamma$ decreases towards zero, there must be a certain negative, real value of $\gamma$, for which D is zero. When $\gamma$ is zero, D will be positive, and equal to $\frac{27\,F\epsilon^2}{4} + F^2\epsilon$. As $D = 0$ is an equation of third degree in $\gamma$, we know that, besides the negative value of $\gamma$, already mentioned, there must be two positive real, or two imaginary, values of $\gamma$, which make $D = 0$.

Increasing $\gamma$ positively, D will, therefore, decrease to a certain minimum, which may or may not be negative, according as the two remaining roots of $D = 0$ are real or imaginary.

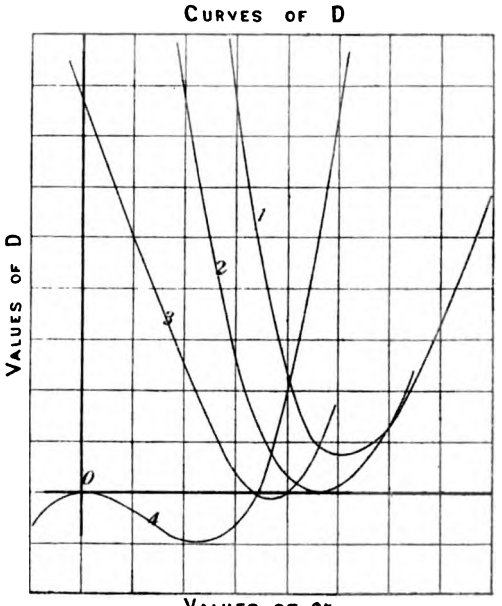
Hereafter, D increases in the positive direction, and when $\gamma > \frac{F}{3}$, in which case A is positive, D can never become negative. Increasing $\gamma$ beyond this, D will increase to $+\infty$. By differentiation of D, we find that the maximum and minimum values of D are given by—

$$\gamma = \frac{F}{12} \pm \sqrt{\frac{F^2}{144} + \frac{3}{2}F\epsilon}$$

showing that D has a minimum on the positive side and a maximum on the negative side.

The general character of the variation of D is best shown by means of curves drawn on values of $\gamma$ as abcissæ.

The diagram, Fig. 6, shows four curves: (1) giving one negative and two imaginary roots of the equation $D = 0$; (2) one negative and two positive equal roots; (3) one negative and two positive unequal roots; and (4) one positive and two zero roots. We shall examine each of these cases separately, Case (4), later on, being a special case.




F IS THE SAME FOR ALL CURVES.
1 . F - 19 ε
2 F - 27 ε
3 : F - 56 ε
4 : ε - 0

Fig. 6.

*Case* (1).

Here D is positive for all positive values of $\gamma$, wherefore the equation for $\lambda$ will have one real root, known to be negative, and two imaginary roots, generally consisting of a real and an imaginary part. Let the three roots be—

$$\lambda_1, \quad \lambda_2 = s + iu, \text{ and } \lambda_3 = s - iu,$$

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therefore—

$$(\lambda - \lambda_1)(\lambda - s - i u)(\lambda - s + i u) = 0;$$

which, taken together with equation (11), gives—

$$\frac{Q_0}{M v} = -(\lambda_1 + 2 s)$$

$$\frac{\gamma}{M \rho^2} = s^2 + u^2 + 2 \lambda_1 s$$

$$\frac{\epsilon Q_0}{M^2 \rho^2 v} = -\lambda_1 (s^2 + u^2);$$

from which—

$$s^3 + s^2 \frac{Q_0}{M v} + \frac{1}{4} s \left[ \frac{Q_0^2}{M^2 v^2} + \frac{\gamma}{M \rho^2} \right] + \frac{Q_0 \beta}{8 M^2 \rho^2 v} = 0,$$

showing that $s = 0$ when $\beta = 0$, *i.e.*, $\gamma = \epsilon$, and for no other real value of $\gamma$. When $\beta$ is negative and small, $s$ becomes positive. When $\beta$ is positive, $s$ will be negative.

If we put—

$$A_2 + A_3 = L_2$$
$$i(A_2 - A_3) = L_3$$
$$B_2 + B_3 = M_2$$
$$i(B_2 - B_3) = M_3;$$

the equations (9) and (10) may be written—

$$\theta = A_1 e^{\lambda_1 t} + e^{s t}[L_2 \cos u t + L_3 \sin u t] + \theta_0 - f \frac{a + z}{\epsilon} \tag{28}$$

$$y = B_0 + B_1 e^{\lambda_1 t} + e^{s t}[M_2 \cos u t + M_3 \sin u t] + \left[ \frac{f}{Q_0} + \frac{f(a + z)}{\epsilon} \right] v t \tag{29}$$

showing that the motion is oscillatory, the period of the oscillation being given by $T = \frac{\pi}{u}$, where T is the time of a complete single swing.

The first exponential term is in both equations evanescent, since $\lambda_1$ is negative. If the centre of lateral resistance falls aft of the centre of gravity, we have seen that $s$ will be negative, and the oscillatory terms will then also ultimately vanish, *i.e.*, the motion will be stable. If, on the other hand, the centre of lateral resistance falls forward of the centre of gravity, $s$ will be positive, and the amplitude of the oscillations will go on increasing, *i.e.*, the motion will be unstable.

In the former case, the motion will tend towards a limiting condition, given by—

$$\theta_1 = \theta_0 - f \frac{a + z}{\epsilon} \tag{30}$$

$$y = B_0 + \left[ \frac{f}{Q_0} + \frac{f(a + z)}{\epsilon} \right] v t \tag{31}$$

giving a limiting vertical velocity

$$\left( \frac{d y}{d t} \right)_1 = \left[ \frac{f}{Q_0} + \frac{f(a + z)}{\epsilon} \right] v. \tag{32}$$

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*Case* (2).

Here the curve of D touches the axis of abscissæ, giving $D = 0$ for two positive equal values of $\gamma$. This is the dividing condition for oscillatory and non-oscillatory motion, and it is, therefore, of interest to know the relations which in this case exist between $\epsilon$ and F.

The equation $D = 0$, having two equal roots at this point, its discriminant $D'$ must here be zero. It will be found that $D' = 0$, when—

$$27^3 \times \epsilon^3 - 3 \times 27^2 F \epsilon^2 + 3 \times 27 F^2 \epsilon - F^3 = 0,$$

and this is satisfied only when $F = 27\,\epsilon$. The roots of the equation $D = 0$ are then found to be—

$$\gamma = 9\,\epsilon, \quad \gamma = 9\,\epsilon, \quad \text{and} \quad \gamma = -\frac{45}{4}\,\epsilon.$$

Thus—

$$\gamma = 9\,\epsilon = \frac{F}{3}$$

is the point where the curve of D touches the axis of abscissæ.

The equation for $\lambda$ gives—

$$\lambda_1 = \lambda_2 = \lambda_3 = -\frac{Q_0}{3\,M\,v}$$

all roots real, negative, and equal.

The equations (9) and (10) become—

$$\theta = e^{-\frac{Q_0}{3\,M\,v}\,t}\left[A_1 + A_2\,t + A_3\,t^2\right] + \theta_0 - f\,\frac{(a+z)}{\epsilon} \qquad (33)$$

$$y = B_0 + e^{-\frac{Q_0}{3\,M\,v}\,t}\left[B_1 + B_2\,t + B_3\,t^2\right] + \left[\frac{f}{Q_0} + f\,\frac{(a+z)}{\epsilon}\right] v\,t. \qquad (34)$$

The motion is non-oscillatory, tending towards the same limiting condition as in Case 1. In this special case we have—

$$F = 3\,\gamma, \qquad \beta = 8\,\epsilon, \qquad \frac{Q_0^2\,\rho^2}{M\,v^2} = 27\,\epsilon,$$

$$\frac{a^2}{\rho^2} = \frac{64}{27}\,\frac{g \times BG}{v^2},$$

where $g$ is the acceleration of gravity.

*Case* (3).

If the expression for $D'$ is differentiated with respect to F, and if, in the result, we put $F = (27 + n)\,\epsilon$, it will be found that—

$$\frac{d\,D'}{d\,F} = -\frac{(27+n)\,\epsilon^2}{27 \times 64}\,(81\,n^2 + 5\,n^3)$$

showing that, if we increase F a little beyond the particular value, $F = 27\,\epsilon$, $D'$ will become negative, showing that the equation $D = 0$ will then have three real roots, $\gamma$. The curve of D will, therefore, cut the axis of abscissæ in two points on the positive side, and between the values of $\gamma$, corresponding to these points, D will be negative. This is Case (3).



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A decrease in F ($n$ negative) makes D′ positive, *i.e.*, one real and two imaginary roots of $D = 0$. This is Case (1).

Thus, the condition that Case (3) shall at all exist is $F > 27\,\epsilon$. If this is not the case, the motion will always be oscillatory; and, if it is the case, and $\gamma$ at the same time has such a value that D is negative, we shall have three real and negative values of the exponents $\lambda$, and the motion will be non-oscillatory.

The equations for $\theta$ and $y$ become—

$$\theta = A_1 e^{\lambda_1 t} + A_2 e^{\lambda_2 t} + A_3 e^{\lambda_3 t} + \theta_0 - f\frac{a+z}{\epsilon} \qquad (35)$$

$$y = B_0 + B_1 e^{\lambda_1 t} + B_2 e^{\lambda_2 t} + B_3 e^{\lambda_3 t} + \left[\frac{f}{Q_0} + \frac{f(a+z)}{\epsilon}\right] v\,t. \qquad (36)$$

The exponential terms are, in this case, always evanescent, and the limiting conditions are the same as in Cases (1) and (2).

In the two special cases where $\gamma$ has such a value that $D = 0$, *i.e.*, the points where the curve of D cuts the axis of abscissæ, we find—

$$p_1 = q_1 = -\sqrt[3]{\frac{B}{2}}$$

$$\lambda_1 = -2\sqrt[3]{\frac{B}{2}} - \frac{Q_0}{3\,M\,v}$$

$$\lambda_2 = \lambda_3 = \sqrt[3]{\frac{B}{2}} - \frac{Q_0}{3\,M\,v}.$$

$$\theta = A_1 e^{\lambda_1 t} + e^{\lambda_2 t}[A_2 + A_3 t] + \theta_0 - f\,\frac{a+z}{\epsilon} \qquad (37)$$

$$y = B_0 + B_1 e^{\lambda_1 t} + e^{\lambda_2 t}[B_2 + B_3 t] + \left[\frac{f}{Q_0} + \frac{f(a+z)}{\epsilon}\right] v\,t. \qquad (38)$$

The motion is here non-oscillatory; but for all other values of $\gamma$, lower or higher than those for which D is zero or negative, the motion will be oscillatory. If $\gamma > \frac{F}{3}$ the motion will always be oscillatory.

It will be found that, with small values of $\epsilon$, the inequality $F > 27\,\epsilon$ may be readily secured, and, probably, also, the value of $\gamma$ necessary to make D negative. But, if $\epsilon$ is very great, we can only make $F > 27\,\epsilon$ by high speed; and it will be found that, with the high value of F, $\gamma$ and, therefore, $\beta$ must have a very high value, in order to make D negative. In other words, regarding it as desirable to make the motion non-oscillatory, the centre of lateral resistance should be drawn farther aft in a boat with great longitudinal stability, than in a boat with small longitudinal stability.

*Limiting Conditions.*

We have seen that whether, or not, the boat in stable motion oscillates, during the first period, after a disturbance has occurred, she will ultimately settle down to a limiting state—

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$$\theta_1 = \theta_0 - \frac{f(a+z)}{\epsilon}$$

$$\left(\frac{dy}{dt}\right)_1 = \frac{f}{Q_0} v + \frac{f(a+z)}{\epsilon} v.$$

The first term of the limiting vertical velocity follows the sign of the disturbing force; the second is of sign opposite to that of its moment about C.

The term $\frac{fz}{\epsilon}$ is the inclination produced by $fz$, if the boat were at rest. This angle is, in the following, denoted by $\phi$ on several occasions.

If $f$ is acting at the centre of lateral resistance, we have—

$$\theta_1 = \theta_0 \text{ and } \left(\frac{dy}{dt}\right)_1 = \frac{f}{Q_0} v;$$

*i.e.*, the boat will settle down to the original inclination, but will acquire a steady vertical velocity.

If $z = -\frac{\gamma}{Q_0}$, we get—

$$\theta_1 = \theta_0 + \frac{f}{Q_0}, \quad \left(\frac{dy}{dt}\right)_1 = 0, \quad \text{and } y_1 = B_0;$$

*i.e.*, the boat suffers a certain angular deflection, but will, in the limit, proceed along a horizontal line, lying a distance $B_0$ from the original level.

If $f$ is acting at G, $z = 0$, and we get—

$$\theta_1 = \theta_0 - \frac{fa}{\epsilon}, \quad \left(\frac{dy}{dt}\right)_1 = \frac{f}{Q_0} v + \frac{fa}{\epsilon} v.$$

This may occur, for instance, if the boat passes through water of varying density.

If an internal weight $f$ is moved through a distance $z$ inside the boat, a pure couple $fz$ is produced. If the weight is moved from forward to aft the limiting conditions become—

$$\theta_1 = \theta_0 - \frac{fz}{\epsilon}$$

and—

$$\left(\frac{dy}{dt}\right)_1 = \frac{fz}{\epsilon} v.$$

Already, when $z = +\frac{\epsilon}{Q_0}$, this disturbance will equal that caused by adding a weight at C, but $z$ may easily exceed this quantity many times. Hence, the shifting of a certain weight is generally more dangerous than the addition or removal of the same weight at the middle of the boat.

*Initial Motion.*

The acceleration is, initially—

$$\frac{d^2\theta}{dt^2} = -\frac{fz}{M\rho^2}$$

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and—

$$\frac{d^2 y}{d t^2} = \frac{f}{M}.$$

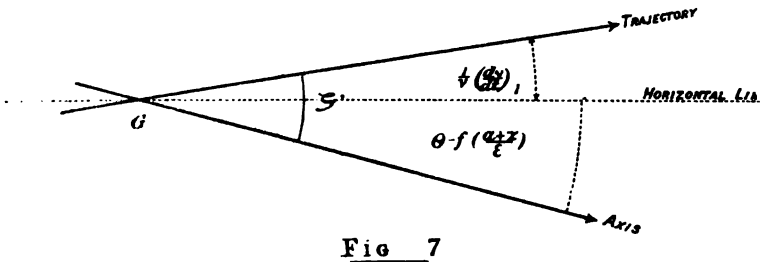


Fig 7

It is of interest to notice that, if $f$ is acting aft, the path of the centre of gravity will at first swerve out to the side opposite to that to which it is ultimately turning. The same phenomenon is observed in the turning circles of ordinary vessels, where the rudder is placed aft. (See Fig. 5.)

*Justification for reckoning the Velocity along the Trajectory equal to Original Velocity, $v$.*

The angle $\zeta$, between the axis of the boat and the tangent to the limiting trajectory of the centre of gravity, is—

$$\zeta = \theta_0 - f\frac{a+z}{\epsilon} + \frac{1}{v}\left(\frac{d y}{d t}\right)_1 = \theta_0 + \frac{f}{Q_0}$$

and differs, therefore, from that of steady motion only by the angle $\frac{f}{Q_0}$, which it appears justifiable to neglect as long as $f$ is small. Thus the resistance, and, therefore, the speed along the trajectory, will remain practically unaltered.

*Effect of Free Liquid Surfaces.*

Suppose a rectangular tank of length $l$ and breadth $b$ to be partly filled with water, then by longitudinal inclination a moment will be created $= \frac{1}{420} b\, l^3 \theta = k\, \theta$ ft. ts.

Working this moment in the equations of motion, we find that the effect is precisely the same as if $\epsilon$ had been reduced by a quantity $k$. The limiting conditions are—

$$\theta_1 = \theta_0 - f\frac{a+z}{\epsilon - k}$$

$$\left(\frac{d y}{d t}\right)_1 = \frac{f}{Q_0} v + f\frac{a+z}{\epsilon - k} v$$

and the exponents are determined by—

$$\lambda^3 + \lambda^2 \frac{Q_0}{M v} + \lambda \frac{\gamma - k}{M \rho^2} + \frac{Q_0 (\epsilon - k)}{M^2 \rho^2 v} = 0.$$

If $k = \epsilon$ the motion becomes unstable.

*Centre of Lateral Resistance Coincides with Centre of Gravity.*

Here $a = 0$, therefore $\beta = 0$ and $\gamma = \epsilon$. The equations of motion become—

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$$\frac{d^2 y}{d t^2} + \frac{Q_0}{M v}\frac{d y}{d t} + \frac{Q_0}{M}\theta = \frac{Q_0 \theta_0 + f}{M}$$

$$\frac{d^2 \theta}{d t^2} + \theta \frac{\epsilon}{M \rho^2} = \frac{\theta_0 \epsilon - f z}{M \rho^2}.$$

In this case $D = \epsilon^3 + 2 F \epsilon^2 + F^2 \epsilon$, which is always positive. The motion will therefore always be oscillatory. The exponents are—

$$\lambda_1 = -\frac{Q_0}{M v}, \quad \lambda_2 = +i\sqrt{\frac{\epsilon}{M \rho^2}}, \quad \lambda_3 = -i\sqrt{\frac{\epsilon}{M \rho^2}}.$$

Moreover—

$$s = 0, \quad A_1 = 0, \quad A_2 = A_3 = \frac{K}{2}, \quad L_2 = K, \quad L_3 = 0.$$

Put—

$$\frac{f z}{\epsilon} = \phi \text{ and } T = \frac{\pi}{\sqrt{\frac{\epsilon}{M \rho^2}}},$$

then the solution of the equations of motion is—

$$\theta - \theta_0 = -\phi\left[1 - \cos\frac{\pi}{T}t\right]$$

$$y = B_0 + B_1 e^{-\frac{Q_0}{M v}t} + M_2 \cos\frac{\pi t}{T} + M_3 \sin\frac{\pi t}{T} + \left[\frac{f}{Q_0} + \phi\right] v t.$$

The angular motion is one of pure unresisted oscillations, the boat oscillating between the positions $\theta_0$ and $\theta_0 - 2\phi$. Ultimately the oscillations would in reality be extinguished by the resistance, varying with angular velocity, and the limiting position would be $\theta_1 = \theta_0 - \phi$.

The vertical motion will be oscillatory, but will ultimately settle down to—

$$\left(\frac{d y}{d t}\right)_1 = \left[\frac{f}{Q_0} + \phi\right] v.$$

*Longitudinal Stability Vanished.*

Here $\epsilon = 0$, and, therefore, $\gamma = \beta$,

$$\theta_0 = \frac{w p}{Q_0 (p - a)} = \frac{S_0 p a_0}{\beta}$$

$$\frac{d^2 \theta}{d t^2} + \frac{\beta}{M \rho^2}\theta + \frac{\beta}{M \rho^2 v}\frac{d y}{d t} = \frac{\theta_0 \beta - f z}{M \rho^2}$$

$$\frac{d^2 y}{d t^2} + \frac{Q_0}{M v}\frac{d y}{d t} + \frac{Q_0}{M}\theta = \frac{Q_0 \theta_0 + f}{M}$$

The solution is—

$$\theta = \theta_0 - \frac{f z}{\beta} + \frac{f (z + a) \rho^2}{M v^2 a^2} - \frac{f (z + a)}{M a v} t + A_2 e^{\lambda_2 t} + A_3 e^{\lambda_3 t}$$

$$y - \frac{f (z + a) \rho^4}{M a^3 v^2} - \frac{f z \rho^2}{\beta a} - \frac{f (z + a) \rho^2}{M v a^2} t + \frac{1}{2}\frac{f (z + a)}{M a} t^2 + B_2 e^{\lambda_2 t} + B_3 e^{\lambda_3 t}$$

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where—

$$\lambda_{2,3} = -\frac{Q_0}{2\,M\,v} \pm \sqrt{\frac{Q_0^2}{4\,M^2\,v^2} - \frac{\beta}{M\,\rho^2}}.$$

As long as—

$$\beta \leqq \frac{Q_0^2\,\rho^2}{4\,M\,v^2},\ i.e.,\ \leqq \frac{F}{4}$$

the roots $\lambda_2$ and $\lambda_3$ will be real, and the motion non-oscillatory. The exponents being negative, the exponential terms will be evanescent. If $\beta > \frac{F}{4}$, the motion will be oscillatory.

Curve (4) on Fig. 6 shows the values of D in this case, corresponding to various values of $\beta$.

Applying the equations to an ordinary ship turning horizontally, we have $\theta_0 = 0$. If the centre of lateral resistance lies aft of the centre of gravity, *i.e.*, $a$ positive (a case which probably occurs in certain torpedo boats), then the equations for $\theta$ and $y$, given above, will hold, if we only put $\theta_0 = 0$, and make $z$ negative.

If we suppose $a = 0$, and, therefore, $\beta = 0$, a case which is very likely to be approached by ordinary ships, we get—

$$\frac{d^2\theta}{d\,t^2} = \frac{f\,z}{M\,\rho^2},$$

$$\frac{d^2 y}{d\,t^2} + \frac{Q_0}{M\,v}\frac{d\,y}{d\,t} + \frac{Q_0}{M}\theta = \frac{f}{M};$$

from which—

$$\theta = \frac{f\,z}{2\,M\,\rho^2}t^2,$$

and—

$$y = \left[\frac{M\,v^2 f}{Q_0^2} - \frac{M^2\,v^4 f\,z}{\rho^2\,Q_0^3}\right]e^{-\frac{Q_0}{M\,v}t} - \frac{f\,z\,v}{6\,M\,\rho^2}t^3 + \frac{f\,z\,v^2}{2\,Q_0\,\rho^2}t^2 + \left[\frac{v\,f}{Q_0} - \frac{v^3\,M\,f\,z}{\rho^2\,Q_0^3}\right]t.$$

These equations hold good, however, only during the first stages of the turning, while $\theta$ is still small.

The limiting trajectory of an ordinary ship is known to be a circle, in which the ship turns with steady velocity, her head turned inwards, the axis forming a certain angle—the "drift angle"—with the tangent to the turning circle.

This case is analogous to the steady limiting motion of the submarine boat, only the trajectory is in the latter a straight line, in the ship a circle. The "drift angle" corresponds to $\zeta$. The force acting on the rudder is: $S_0 \sin a_0 \cos a_0 = f$. Let $r$ be the radius of the turning circle, then one equation of motion will be—

$$\frac{M\,v^2}{r} = Q_0\,\zeta - f$$

or—

$$\zeta = \frac{\frac{M\,v^2}{r} + f}{Q_0} = \text{drift angle.}$$

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The turning moment is balanced by the resistance, which may be taken in this case approximately to be $W\,k\left(\frac{d\theta}{dt}\right)^2$.* The angular acceleration is zero, and the angular velocity therefore constant, let it be $\omega$.

Thus—

$$f\,p - W\,k\,\omega^2 - Q_0\,\zeta\,a = 0$$

or—

$$k\,W\,\omega^2 = f\,(p - a) - \frac{M\,a\,v^2}{r}.$$

Not knowing the law according to which $Q_0$, $k$, and $a$ vary, these formulæ can, however, hardly be turned to any practical use in the present state of our knowledge.

*Impulsive Forces.*

Let an upward impulsive force P be acting forward, and at a distance $z$ from G.

Let the impulse create an angular velocity $\omega$ and a vertical velocity $v_1$, then—

$$P = M\,v_1 \text{ and } \omega = -\frac{P\,z}{M\,\rho^2}$$

or—

$$v_1 = -\,\omega\,\frac{\rho^2}{z}.$$

The equations become—

$$\theta = A_1\,e^{\lambda_1 t} + A_2\,e^{\lambda_2 t} + A_3\,e^{\lambda_3 t} + \theta_0$$

$$y = B_0 + B_1\,e^{\lambda_1 t} + B_2\,e^{\lambda_2 t} + B_3\,e^{\lambda_3 t}$$

the motion being oscillatory, or not, according to the value of D.

With the altered initial conditions, we get—

$$A_1 + A_2 + A_3 = 0$$

$$A_1\,\lambda_1 + A_2\,\lambda_2 + A_3\,\lambda_3 = -\,\omega$$

$$B_0 + B_1 + B_2 + B_3 = 0$$

$$A_1\,\lambda_1^2 + A_2\,\lambda_2^2 + A_3\,\lambda_3^2 = \frac{\beta\,v_1}{M\,\rho^2\,v} = L,$$

from which—

$$A_1 = \frac{L + \omega\,(\lambda_2 + \lambda_3)}{(\lambda_1 - \lambda_3)\,(\lambda_1 - \lambda_2)}$$

and similarly for $A_2$ and $A_3$.

The limiting conditions are—

$$\theta_1 = \theta_0, \qquad \left(\frac{d\,y}{d\,t}\right)_1 = 0, \qquad y_1 = B_0 = \frac{P\,z\,v}{\beta}.$$

* Manual of Naval Architecture, by W. H. White, 1882, page 615.

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*Pure Unresisted Oscillations.*

If the disturbing force is impulsive the differential equations are—

$$\frac{d^2\theta}{dt^2} + \frac{\gamma}{M\rho^2}\theta = \frac{\gamma\theta_0}{M\rho^2}$$

$$\frac{d^2y}{dt^2} + \frac{Q_0}{M}\theta = \frac{Q_0\theta_0}{M}.$$

The general solution of the first equation is—

$$\theta = L_1 \cos t\sqrt{\frac{\gamma}{M\rho^2}} + L_2 \sin t\sqrt{\frac{\gamma}{M\rho^2}} + \theta_0.$$

Suppose now that at time $t = 0$ the boat has received an upward impulse acting aft, which has created a vertical velocity—

$$v_1 = \frac{Q_0\phi T}{M\pi}$$

and an angular velocity—

$$\omega = \frac{\pi\phi}{T}$$

where—

$$T = \frac{\pi}{\sqrt{\frac{\gamma}{M\rho^2}}}$$

We then find—

$$\theta - \theta_0 = \phi \sin\frac{\pi t}{T}$$

$$y = \frac{\phi Q_0 T^2}{M\pi^2}\sin\frac{\pi t}{T}.$$

Thus the path of G is a curve of sines oscillating on both sides of the original level, and with a half amplitude—

$$y_1 = \pm\frac{\phi Q_0 T^2}{M\pi^2}.$$

The horizontal length of the oscillations will be—

$$L = Tv = \pi v\sqrt{\frac{M\rho^2}{\gamma}}.$$

Angularly the boat will oscillate about the original position $\theta_0$ with a half amplitude equal to $\pm\phi$.

It is found that, when the boat is at its lowest level, the axis will have its minimum inclination to the horizon, and *vice versâ*, the consequence of which is, that, when the boat is ascending, the fore-end swings downwards, and when descending it swings upwards.

Let us now imagine lateral resistance to act. The force Q will be subject to fluctuations, and will, apart from the effect of angular motion, be augmented during ascent, and diminished during

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descent. If the centre of lateral resistance C lies aft of G, the effect will be that the angular motion is always opposed by these fluctuations. This is expressed in the formulæ for resisted motion by the fact that $s$ is negative. Conversely, the fluctuations in Q will assist and augment the angular motion, if C lies forward of G; and $s$ is, in this case, positive.

The fluctuations in Q will, moreover, cause an oscillatory motion of G, which will combine with that due to angular motion. The two oscillatory motions will mutually react on each other, and the result will be a curve determined by two terms, one containing a sine, the other a cosine.

*Resistance Varying as Angular Velocity Included.*

The differential equations become—

$$\frac{d^2\theta}{dt^2} + \frac{k}{M\rho^2}\frac{d\theta}{dt} + \frac{\beta}{M\rho^2 v}\frac{dy}{dt} + \frac{\gamma}{M\rho^2}\theta = \frac{\theta_0\gamma - fz}{M\rho^2}$$

$$\frac{d^2y}{dt^2} + \frac{Q_0}{Mv}\frac{dy}{dt} + \frac{Q_0}{M}\theta = \frac{\theta_0 Q_0 + f}{M}$$

The solution of these equations is found to be of the same form as when this resistance was not included, but the exponents are given by—

$$\lambda^3 + \lambda^2\left[\frac{Q_0}{Mv} + \frac{k}{M\rho^2}\right] + \lambda\left[\frac{\gamma}{M\rho^2} + \frac{Q_0 k}{M^2\rho^2 v}\right] + \frac{\epsilon Q_0}{M^2\rho^2 v} = 0.$$

It is seen from this equation that $s$ will be zero for a value of $\gamma$ which is somewhat smaller than $\epsilon$; in other words, the oscillatory terms may be evanescent, even if the centre of lateral resistance lies a little forward of the centre of gravity.

*Stability of Motion.*

Suppose the boat deflected a small angle $\phi$ from the position of equilibrium of steady motion, and left there without any angular velocity. If she now returns to her original position, the motion must be stable.

The solution of the equations of disturbed motion will be of the form—

$$\theta = \theta_0 + A_1 e^{\lambda_1 t} + A_2 e^{\lambda_2 t} + A_3 e^{\lambda_3 t}$$

$$y = B_0 + B_1 e^{\lambda_1 t} + B_2 e^{\lambda_2 t} + B_3 e^{\lambda_3 t}$$

and the constants are found by the same formula as above, only $L = -\frac{\varphi\gamma}{M\rho^2}$ and $K \doteq \phi$.

If, now, $\beta$ and $\epsilon$ are both positive, we know that all the exponential terms will be evanescent, and we get ultimately: $\theta_1 = \theta_0$ and $y_1 = B_0$, *i.e.*, the boat returns to its original inclination, and the motion is stable.

If $\epsilon = 0$, $\beta$ being still positive, we find $A_1 = 0$ and $B_1 = 0$, and, $\lambda_2$ and $\lambda_3$ being negative, the motion will still be stable.

If $\epsilon$ is negative, and $\beta$ positive, there will be at least one positive value of $\lambda$, and the motion will be ever increasing, and therefore unstable.



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If $\beta = 0$, and $\epsilon$ positive, we find $A_1 = 0$, $L_3 = 0$, $L_2 = \phi$, and, therefore—

$$\theta = \theta_0 + \phi \cos \frac{\pi}{T} t$$

$$y = B_0 + B_1 e^{-\frac{Q_0}{M v} t} + M_2 \cos \frac{\pi}{T} t + M_3 \sin \frac{\pi}{T} t,$$

showing that the boat will continue to oscillate indefinitely. The resistance to angular motion will, however, in reality extinguish the oscillations, and we may, therefore, still consider this as a case of stable motion.

If $\beta$ is negative and numerically $< \epsilon$, which is positive, $\gamma$ will still be positive, and no positive real value can satisfy the equation for $\lambda$, but there will be two imaginary roots having positive real parts, which shows the motion to be oscillatory, ever increasing, and therefore unstable.

This conclusion must, indeed, be somewhat modified, because of the resistance to angular motion, which will permit $\beta$ to have a small negative value before the motion becomes unstable.

The general conditions of stability of motion are accordingly $\epsilon \geqq 0$, and $\beta \geqq 0$, and $\gamma > 0$.

*Manœuvring Power.*

We have—

$$\frac{d^2 \theta}{d t^2} = \frac{S_0 p a - \left[ (\epsilon + \beta) \theta + \frac{\beta}{v} \frac{d y}{d t} + k \frac{d \theta}{d t} \right]}{M \rho^2},$$

or—

$$\frac{d^2 \theta}{d t^2} = \frac{\text{Turning moment of rudder} - [\text{stability of motion} + \text{moment of resistance}]}{\text{moment of inertia}}$$

which determines the rate at which angular velocity is acquired or extinguished.

Neither stability of motion nor resistance should be sacrified, in order to attain manœuvring power, which should be secured exclusively through the turning moment of the rudder.

*Surplus Buoyancy.*

In steady motion, when the conning tower was placed vertically over the centre of gravity, we found—

$$\theta_0 = \frac{w \rho}{\delta} \quad \text{and} \quad a_0 = \frac{w \gamma}{S_0 \delta}.$$

Suppose, now, the conning tower to be placed forward, or aft, of G, at a distance from this point. It is supposed, always, that, when the boat is at rest, she will float on even keel at the surface, with only the top of the conning tower projecting above the water. The equations of equilibrium of motion become—

$$S_0 p a_0 - \theta_0 \gamma \mp w z = 0,$$
$$S_0 a_0 + w - \theta_0 Q_0 = 0,$$

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from which—

$$\frac{\theta_0}{w} = \frac{(p \pm z)}{Q_0 (p - a) - \varepsilon}.$$

These equations hold for rudder placed aft. If, now, rudder is placed forward, we get—

$$\frac{\theta_0}{w} = \frac{(p \mp z)}{Q_0 (p + a) + \varepsilon}.$$

the upper sign in all equations corresponding to conning tower placed forward, and conversely.

It appears that for obtaining a small value of $\theta_0$, with a great value of $w$, the conning tower should be placed on same side of centre of gravity as the rudder.

Supposing the conning tower thus placed, it remains to examine whether the forward or the aft position is the best.

The formula is—

$$\frac{\theta_0}{w} = \frac{(p - z)}{Q_0 (p \mp a) \mp \varepsilon}$$

upper sign holding for rudder placed aft, lower sign for rudder placed forward. The formula shows that it is most advantageous to place the rudder forward.

In the extreme case when the surplus buoyancy acts directly over the rudder, whether forward or aft, we get $\theta_0 = 0$, *i.e.*, the boat moves on an even keel, and $a_0 = \mp \frac{w}{S_0}$, *i.e.*, turned upwards or downwards in case of an aft or forward rudder respectively.

*The Rudders.*

Any two rudders placed symmetrically with common turning axis are treated as one rudder.

We have seen that it is unfavourable to surplus buoyancy to place the rudder aft, but the bad effect in this respect may be counteracted by placing the conning tower more aft.

Placing the rudder aft increases the stability of motion, but decreases thereby the manœuvring power; further, the leverage of the rudder is greater aft than forward. The difference in turning moment in the two cases will be—

$$S_0 [\theta (p_1 + p_2) - a (p_2 - p_1)],$$

if $p_1$ and $p_2$ are the values of $p$, when the rudder is placed forward and aft respectively. $a$ being much greater than $\theta$, the gain, if any, cannot be great by changing the rudder from aft to forward, and the manœuvring power will, therefore, not be materially affected.

The rate at which a small variation in the rudder angle influences the angle of inclination of the boat is, for any given position of equilibrium, determined by the value of $\frac{d\theta}{da}$. By differentiating the equation of turning moment, we find—

$$\frac{d\theta}{da} = -\frac{S^0 p}{\gamma}$$

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showing the efficacy of the rudder to be directly as the moment of rudder pressure about G, and inversely as the stability of motion.

The effect of the rudder, as regards the limiting condition of the boat, must be the same as that of a disturbing force $S_0 (a - a_0)$, acting at a distance $p$ from G; and the rudder will, consequently, be more effective forward than aft.

When a disturbance of a permanent nature occurs, the instruments will soon indicate that the inclination and the depth of immersion are changed, and the helmsman will, consequently, attempt to bring the boat back to the original depth of immersion, and to find the new position of equilibrium of the boat $\theta_2$, as well as the corresponding rudder angle $a_2$.

These are given by the formula—

$$S_0 p a_2 - \gamma \theta_2 - f z = 0$$

$$S_0 a_2 - Q_0 \theta + w + f = 0$$

from which—

$$\theta_2 = \frac{f(p + z) + p w}{\delta}$$

$$a_2 = \frac{f z Q_0 + (w + f) \gamma}{S_0 \delta}.$$

It is clear that the task of the helmsman must be more difficult in a submarine boat than in an ordinary ship, where disturbances analogous to those here treated do not occur, and where, consequently, the rudder is always carried at practically the same angle after, as before, a deviation from the right course has taken place.

It is recommended in the first part of this paper, to place the rudder, and consequently also the conning tower, aft (*i.e.*, somewhat aft of G), mainly for practical reasons. We have then ($z$ refers here to the conning tower)—

$$\theta_0 = \frac{S_0 p a_0 + w z}{\gamma} = \frac{w(p - z)}{\delta}$$

and—

$$a_0 = \frac{w[Q_0 (a - z) + \epsilon]}{S_0 \delta}.$$

This change in the value of $\theta_0$ and $a_0$ would not, however, affect the form of any of the previous equations.

*Non-Symmetry.*

Imagine a symmetrical boat, altered so as to become unsymmetrical about the horizontal central plane, or the propeller shaft placed away from the axis of the boat, parallel or inclined to the same.

Apart from the system of forces acting on the boat in steady motion, when symmetrical, we shall now find extra forces acting, which are all equivalent to one single force S in the central vertical plane, cutting the axis at some point D. Let the vertical resultant of this force be F, supposed acting downwards.

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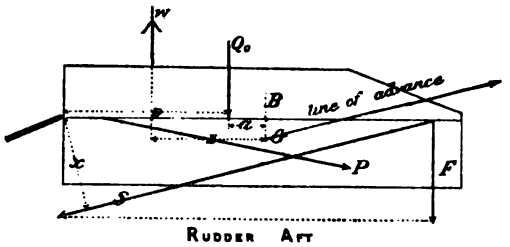
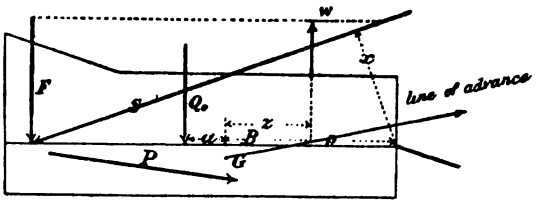



RUDDER AFT

FIG 8




RUDDER FORWARD.

FIG. 9.

Resolving vertically—

$$S_0 a_0 - F + w - Q_0 \theta_0 = 0.$$

Let $x$ be the distance of S from the rudder stock, and take moments about the same—

$$[Q_0 (p \mp a) \mp \epsilon] \; \theta_0 = w (p \mp z) - S x$$

where upper sign holds for an aft-rudder, lower sign for a fore-rudder.

The first equation shows, that for permitting a great surplus buoyancy with small angle of inclination, F should be acting downwards.

The second equation shows that, for the same reason, the moment of S about the rudder stock should always be of opposite sign to the moment of surplus buoyancy, and should be numerically great.

It is readily seen, from Figs. 8 and 9, that these claims are satisfied thus:—In case of an aft rudder (see Diagram, Fig. 8) the lines of the hull should be rather full in the bottom, and cut away forward in the upper part. The propeller shaft should be placed above the axis, or, preferably, inclined downwards as indicated.

In case of a fore-rudder (see Diagram, Fig. 9) the dispositions should be the converse, except that the propeller shaft should also here be inclined downwards if inclined at all.

It is unavoidable that S should change both in magnitude and direction with the speed, and probably also with the depth of immersion, and it is probable, therefore, that it will cause disturbances in steering. Non-symmetry should therefore be avoided, at any rate in experimental boats.

*Heeling.*

If the boat is heeled a small angle $\psi$, and the vertical rudder is laid at an angle $a$, then if $S_0 a$ is the pressure on the rudder when the boat is upright, the pressure will now have a vertical component $S_0 a \psi$ approximately. This force will be equivalent to the action of a horizontal rudder of same area as the vertical rudder, and laid at an angle $a \psi$.

*On Even Keel.*

If surplus buoyancy is to be preserved on even keel, and, if not acting just above the rudder, special means must be provided to produce a permanent downhaul force. This may be created either by inclined planes or rudders, or by vertical propellers.

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The question of inclined planes has been discussed already in the section on rudders, where midship rudders and rudders at both ends are mentioned.

If, in the latter case, it is desired that the four rudders should, in steady motion, all be carried at the same angle, we must have—

$$(S_{01} + S_{02})\, a_0 = w$$

and—

$$S_{01}\, a_0\, p_1 = S_{02}\, a_0\, p_2$$

*i.e.*, the areas of the rudders must be inversely as their distances from the centre of gravity.

The equations for disturbed motion and their solution will be the same as if the boat were inclined, only $\theta_0^{\cdot} = 0$.

Downhaul propellers entail a great expenditure of the very limited store of energy which the boat possesses for submarine propulsion, and they take up much valuable space. They must be generally used in conjunction with a horizontal rudder, by means of which the boat is kept strictly horizontal.

Rising or sinking is effected by varying the speed of the propellers.

The service must in such case be complicated, because both rudder and propellers have to be manœuvred at the same time.

If surplus buoyancy is not to be preserved, the vertical movements may be effected by means of a pump, which produces variations in displacement. The action of such pump is not so quickly reversed as in case of rudders or propellers.

This method has been used by many earlier inventors for keeping the depth under way, and, as it appears, never with success, at any rate not in large boats (*Le Plongeur*). It presents the advantage which will probably make such a pump indispensable in any boat not having downhaul propellers, that it can be used when the boat is at rest (*Le Goubet*).

By all three methods here mentioned the boat must, when rising or sinking, be moved in the direction of her greatest resistance; while by inclining the boat, it can always be moved in direction of its least resistance or nearly so.

### Numerical Examples.

The following data are taken from a design worked out by the author.

Given—

$W = 100$ tons. $v = 6$ knots $= 10$ ft. per second.
Area of rudder $= 20$ sq. ft. $\upsilon = 34$ ft.
Area of central horizontal section $= 700$ sq. ft.
$BG = \cdot 75$ ft. $a = 4\cdot 88$ ft. $w = \cdot 5$ tons. $\rho = 20$ ft.

Assume the lateral resistance given by the formula $Q_0 = \Sigma\,(k\,A\,v^2)$, and assume $k$ in case of the hull to be $= 1$, and in case of the rudder to be $= 2\cdot 25$, then—

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$$Q_0 = \frac{1{\cdot}00 \times 700 \times \overline{10}^2 + 2{\cdot}25 \times 20 \times \overline{10}^2}{2240} = 33 \text{ tons}$$

and—

$$S_0 = \frac{2{\cdot}25 \times 20 \times 10^2}{2240} = 2 \text{ tons.}$$

Moreover we find—

$\epsilon = 75$ ft. tons. $\beta = 161$ ft. tons, and $\gamma = 236$ ft. tons. $M = 3{\cdot}125$. $M\,v = 31{\cdot}25$. $M\,\rho^2 = 1250$.
$\delta = 886$. $\theta_0 = {\cdot}0192$, about 1°. $\alpha_0 = {\cdot}0666$, about 4°. $F = 1394$. $F = 18{\cdot}6\,\epsilon$,

wherefore the motion will be oscillatory for all values of $\gamma$.

(1) *A downwards Force acting forward.*

Given—

$$f = -{\cdot}075 \text{ tons.} \qquad z = +30 \text{ ft.} \qquad f\,z = 2{\cdot}25 \text{ ft. tons,}$$

the equation for the exponents is—

$$\lambda^3 + \lambda^2\,1{\cdot}056 + \lambda\,{\cdot}1888 + {\cdot}0634 = 0,$$

which gives—

$$A = -{\cdot}18290. \qquad B = +{\cdot}08413. \qquad \frac{B^2}{4} + \frac{A^3}{27} = +{\cdot}001542.$$

$$\left.\begin{matrix} p_1^3 \\ q_1^3 \end{matrix}\right\} = -{\cdot}04207 \pm \sqrt{{\cdot}001542} = \left\{\begin{matrix} -{\cdot}00280 \\ -{\cdot}08134 \end{matrix}\right.$$

$$p_1 = -{\cdot}1408. \qquad q_1 = -{\cdot}4333.$$
$$\lambda_1 = -{\cdot}1408 - {\cdot}4333 - {\cdot}3520 = -{\cdot}9261.$$

$$\left.\begin{matrix} p_2 \\ p_3 \end{matrix}\right\} = {\cdot}0704 \mp 1220i. \qquad \left.\begin{matrix} q_2 \\ q_3 \end{matrix}\right\} = {\cdot}2166 \mp {\cdot}3752i.$$

$$\lambda_2 = -{\cdot}0649 + {\cdot}2532i. \qquad s = -{\cdot}0649$$
$$\lambda_3 = -{\cdot}0649 - {\cdot}2532i. \qquad u = +{\cdot}2532.$$

$$A_0 = +{\cdot}0192 + {\cdot}0349 = +{\cdot}0541. \qquad L = +{\cdot}0018. \qquad K = -{\cdot}0349.$$
$$A_1 = -{\cdot}00072. \qquad L_2 = -{\cdot}0356. \qquad L_3 = -{\cdot}0118.$$
$$B_1 = -{\cdot}0632.$$

$$\frac{A_1}{\lambda_1} + \frac{A_2}{\lambda_2} + \frac{A_3}{\lambda_3} = +{\cdot}0812. \qquad B_0 = +1{\cdot}1898.$$

$$M_2 = -1{\cdot}1267. \qquad M_3 = +{\cdot}9472.$$

$$+\left[\frac{f}{Q_0} + \frac{f\,(a+z)}{\epsilon}\right] v = -{\cdot}3715.$$

The expressions for $\theta$ and $y$ are:

$$\theta = -{\cdot}00072\,e^{-{\cdot}9261\,t} - e^{-{\cdot}0649\,t}\,[{\cdot}0356 \cos({\cdot}2532\,t) + {\cdot}0118 \sin({\cdot}2532\,t)] + {\cdot}0541.$$
$$y = -{\cdot}0632\,e^{-{\cdot}9261\,t} - e^{-{\cdot}0649\,t}\,[1{\cdot}1267 \cos({\cdot}2532\,t) - {\cdot}9472 \sin({\cdot}2532\,t)] - {\cdot}3715\,t + 1{\cdot}1898.$$

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The limiting conditions are—

$$\theta_1 = +\cdot 0541, \text{ about } 3^\circ \text{ inclination to the horizon downwards.}$$

$$\left(\frac{dy}{dt}\right)_1 = \cdot 3715 \text{ ft. per second downwards, or about 22 ft. per minute.}$$

The steepness of the trajectory is $\frac{1}{v}\left(\frac{dy}{dt}\right)_1 = \cdot 03715$, about 2° inclined to the horizon. The angle between the axis and the trajectory is $\zeta = \cdot 0541 - \cdot 0372 = \cdot 0169$, about 1°, which is the drift angle in the limiting disturbed motion. $\zeta$ differs from the drift angle in steady motion, $\theta_0$, only by the small amount: $\cdot 0192 - \cdot 0169 = \cdot 0023 = \frac{f}{Q_0}$, about 8′. The resistance, and consequently the speed along the trajectory, must therefore be practically the same in both cases.

Before coming to rest, the boat will oscillate with a period—

$$T = \frac{\pi}{\cdot 2532} = 12\cdot 4 \text{ seconds.}$$

The curve 1 on Plate XLVII. shows the path of the centre of gravity in this case. The ordinates are on a larger scale than the abscissæ in order to show the undulations more clearly.

(2) *A downward Force acting Aft.*

Let $z = -30$ ft., as before $f = -\cdot 075$ tons.

The values of $\lambda_1$, $s$, and $u$ remain unaltered.

$$\frac{f(a+z)}{\epsilon} = -\cdot 0251. \qquad A_0 = -\cdot 0059.$$

$$L = -\cdot 0018. \qquad K = \cdot 0251. \qquad A_1 = -\cdot 00010.$$

$$L_2 = +\cdot 0252. \qquad L_3 = +\cdot 0061. \qquad B_1 = -\cdot 0088.$$

$$\frac{A_1}{\lambda_1} + \frac{A_2}{\lambda_2} + \frac{A_3}{\lambda_3} = -\cdot 0464. \qquad B_0 = -\cdot 6804.$$

$$M_2 = +\cdot 6892. \qquad M_3 = -\cdot 7577. \qquad \left[\frac{f}{Q_0} + \frac{f(a+z)}{\epsilon}\right] v = +\cdot 2285.$$

The expressions for $\theta$ and $y$ become—

$$\theta = -\cdot 0001\, e^{-\cdot 9261\, t} + e^{-\cdot 0649\, t}\left[\cdot 0252 \cos(\cdot 2532\, t) + \cdot 0061 \sin(\cdot 2532\, t)\right] - \cdot 0059.$$

$$y = -\cdot 0088\, e^{-\cdot 9261\, t} + e^{-\cdot 0649\, t}\left[\cdot 6892 \cos(\cdot 2532\, t) - \cdot 7577 \sin(\cdot 2532\, t)\right] + \cdot 2285\, t - \cdot 6804.$$

The limiting conditions are—

$$\theta_1 = -\cdot 0059, \text{ about } \tfrac{1}{3}^\circ \text{ upwards.}$$

$$\left(\frac{dy}{dt}\right)_1 = \cdot 2285 \text{ ft. per second upwards, or about 13 ft. per minute.}$$

The steepness of the trajectory is in the limit smaller than when the weight was acting forward, being now only—

$$\frac{1}{v}\left(\frac{dy}{dt}\right)_1 = \cdot 02285, \text{ about } 1\tfrac{1}{2}^\circ \text{ inclined to the horizon.}$$

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The period of oscillations will be the same.

It will be noticed on Plate XLVII., curve 2, that the path of the centre of gravity falls at first a little below the horizontal line, and then it crosses this line, ascends, and ends in a straight line after some undulations.

The following Table shows the limiting conditions in various cases :—

| Disturbance. | Angular Deflection. | Change of Depth of Immersion in One Minute. |
|---|---|---|
| ·075 tons added 30 ft. forward of G ... ... | 2° downwards. | 22 ft. downwards. |
| ·075 tons added 30 ft. aft of G ... ... ... | $1\frac{1}{3}$° upwards. | 13 ft. upwards. |
| ·075 tons added at G ... ... ... ... | $\frac{1}{2}$° downwards. | 4 ft. downwards. |
| ·075 tons added at centre of lateral resistance ... | 0 | 1 ft. downwards. |
| ·075 tons moved through 60 ft. from aft to forward | $3\frac{1}{2}$° downwards. | 36 ft. downwards. |

## DISCUSSION.

Admiral Sir Nathaniel Bowden-Smith, K.C.B. (Associate): My Lord and Gentlemen, I have some diffidence in speaking before an audience like the present, because I feel that I, being no expert, and having little or no scientific knowledge, am addressing a company of experts; but I am encouraged to speak, because I remember some years ago Sir William White was good enough to say that you liked occasionally to hear the opinions of naval officers. I am very glad to have this opportunity, on behalf of my brother officers, of thanking Captain Hovgaard for coming over here and reading a paper on submarine boats, and to say how glad we are to welcome a naval officer from another country to our shores. The subject is of particular interest to us at present, because, as you know, we have ordered a few of these boats for trial and experiments. I do not propose to deal with what I call the scientific or the learned part of this paper, but will merely offer a few remarks upon submarine boats in general. Now, I would call your attention to the fact that, although the submarine boat in its present advanced state is a modern engine of war, the question of submarine boats themselves is not so very new. People have been trying to get under water for many years. There seems to have been a sort of fascination on the part of human beings to attempt to navigate beneath the surface, as well as to sail on it. In the seventeenth century—in 1640 I think it was—some man tried to invent a submarine boat, and in 1674 a native of Plymouth astonished his fellow countrymen by suggesting a boat, which, he said, was to go under water and come up to the top again at pleasure. He built his vessel, he got his audience, and he sank the boat. She rose to the surface again; but, unfortunately, when she went down a second



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time she failed to come up once more, so that inventor was cut off prematurely. Many of us here will remember the attempt to use submarine boats in the great American civil war. We can call to mind that, when the Federals were blockading Charleston, the people there invented a submarine boat which was said to have drowned three crews. After that she did manage to sink one of the Federal blockading cruisers; I think the *Housatonic*, a ship of 1,200 tons. But in sinking that vessel she again went to the bottom herself, and drowned a fourth crew. We have made great advances since then; but, among all the different types of boats, and all the different inventors who have built these boats, I note that my favourite submarine boat builder has not been mentioned. Perhaps some of you will guess that I allude to that wonderful naval architect, M. Jules Verne. I dare say you will be amused at my mentioning him, but if any of you have not read "Twenty Thousand Leagues Under the Sea," I advise you to do so. I am sure you will be amused as well as interested. However, to speak seriously on this question. In his paper, Captain Hovgaard has said that these boats have only been navigated for about two miles under the water. I was under the impression they could go beneath the surface for a greater distance. It is claimed for the French boat *Narval* that she can be steamed on the surface for 600 miles. I also notice that Admiral O'Neill, the United States Admiral, in conducting some experiments with submarine boats, does not speak so very favourably of the *Holland* type, the one we are adopting for trial. However, he does not appear to be favourably impressed with any of them. As far as I am personally concerned, I cannot say I am very much bitten with these submarine boats, but at the same time I consider that, as the French have four of them in constant practice, and have ordered eight or ten more to be constructed, and, as I understand, the Americans and some other nations are experimenting with them, I am glad that our Admiralty have also ordered some for trial. I do not think we, as a great sea Power, can afford to overlook any new marine invention; and it is desirable that we should know what these boats can do and what they cannot do. After being practically acquainted with them, it will, at all events, enable us better to devise some means of frustrating their attack. I myself should have been contented with three instead of five. However, as the Admiralty has made such a great plunge with boilers, and has put Bellevilles into between sixty and seventy of our ships, I feel relieved that we have not on this occasion made an enormous dive, and ordered sixty submarine boats.

Admiral the Hon. Sir Edmund R. Fremantle, G.C.B., C.M.G. (Associate): My Lord and Gentlemen, I was in hopes that some of the naval architects would have risen to tackle this question from the scientific point of view. I can only concur with my friend, Sir Nathaniel Bowden-Smith, in thanking the author for the very interesting lecture he has given, and make some similar observations to his own. I confess I can be no great believer in submarine boats: nevertheless, I think it is quite right to try them. They have always struck me as something like Mahomet's coffin, which is supposed to be somewhere between heaven and earth. One has to arrange a submarine boat, which has naturally very little surplus buoyancy, to navigate between the bottom and the top of the water. I was rather struck by what has been said about their sliding along the bottom. It would be a new mode of navigation, to us at all events, and to some extent a new idea. There may be something in it, with a suitable bottom. If the submarine boat emulate the flat fish, and so swings along the bottom of the sea, whether over mud or whether over sand, there would be a way of preserving the course, an operation which does not seem quite so simple for these vessels when they act in what is ordinarily supposed to be their function as submarine boats. I have only further to say that I am extremely glad that these vessels should be experimented upon, and their

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merits determined by those who are capable of arriving at a correct conclusion; but at the present moment, at all events, we must not put too much faith in them, notwithstanding the periscope, of which we read in the French papers, or believe that they can be navigated in what is worse than a dense fog. We do not like a fog on board ship. We know it is the greatest danger we can have, and we do not like extreme darkness where no lights are to be seen, and that will be the function of the submarine boats when they have to act. They must in action be under water, and I cannot help thinking there will be extreme difficulty in their so acting. Of course if they can be easily handled with their conning tower above water, so as to steam along showing very little surface, and then be able to navigate with full accuracy under water by means of their two horizontal rudders, and in other ways, so that they can maintain their depth with certainty, they would be of considerable value. I think that the Admiralty are wise, as I have said before, in trying the submarines. I think that whatever is possessed by any foreign nation in the way of naval offence or defence ought to be tried in this country; seeing we depend so entirely upon our naval power. But I confess that I am rather sorry, or perhaps I am rather relieved, like my friend Sir Nathaniel, to find that we have not jumped to one hundred or so, and thus committed ourselves rather too deeply to a particular pattern of submarine boat. At the same time, I do not quite see why we should have ordered so many exactly similar. I cannot help thinking that if some of our naval constructors, and some of our large shipbuilding firms, like Vickers and Armstrong and others, were asked whether they could construct a submarine boat, that we might have some variety, and possibly might have something better than the *Holland*. However, I am glad, as I said before, that we are about to try it. I am quite sure our young officers will be delighted, and perhaps some of our older ones too, to go under water to see what it is like, and how these craft can be handled. If it is true that the French have managed to navigate their submarine boats as well as they say, I am quite sure that the younger officers of the British Navy will be able to emulate them, and that whatever their experience is, if it is properly reported, as undoubtedly it will be to the Admiralty, it will be of great service to this country.

Mr. Sydney W. Barnaby (Member of Council): My Lord and Gentlemen, I rise to speak, not as having any special knowledge of submarine boats; but, as an old friend of Captain Hovgaard, I should just like to say what are his qualifications for reading a paper on this subject. I know he has been studying it for a great many years. When he was a student at the Royal Naval College at Greenwich he spent his spare time in designing a submarine boat there. He has since presented two papers to the Institution, and during his service with his own Government he has still continued to study this question. I think we must all acknowledge that he has given some very interesting and novel information in his paper; at any rate, it seems to me that some of it is novel. I had not realised that the shape of the submarine boat was so important as it seems to be. Usually one finds them of the most elementary form—cigar-shaped; whereas Captain Hovgaard has shown that there is some advantage in modifying that shape, in order to get the centre of lateral resistance well aft. The difficulty with submarine boats, I think, is largely one of detail, and, if we can get the effects of different details, and of changes in detail, clearly put before us, it must assist those who are occupied in studying the subject. Especially do I think the details of both horizontal and vertical steering are important. Captain Hovgaard has shown what a great effect the transference of the weight of one man from one end of the ship to the other has. The weight of one man causes it to descend 36 ft. in one minute—imagine the difficulty of keeping the horizontal direction. You may be at the bottom in a

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minute. You may be at the top in considerably less than a minute. I heard of an interesting and curious incident illustrating this difficulty, which took place in an Italian submarine boat. She was making an under-water run in the harbour at Spezzia, when her crew found that the vessel was suddenly arrested; they could neither go ahead nor astern; they did not know where they were. They pumped out the water from the ballast tanks to increase the boat's flotation, but without effect. They then proceeded, with immense labour, to detach the heavy keel, which was bolted to the bottom outside in order to be let go in case of emergency, but again without effect. They practically gave themselves up for lost. There was a guardship in the harbour of Spezzia, the *Maria Pia*, she is usually anchored there ahead and astern. The captain of this guard ship had heard for some time an extraordinary tapping and scraping underneath the ship's bottom. He could not conceive what was the matter. At last it occurred to him to signal to the arsenal at Spezzia, inquiring if the submarine boat was out that day. The reply came that she was. Then he thought he knew what was the tapping and scraping under the guardship's bottom. He slipped his moorings; he swung the ship round; the submarine boat came up like a cork, and they pulled the men out, more dead than alive. That only illustrates one of the advantages of studying every detail carefully, as Captain Hovgaard has done. I think we are much indebted to him for his interesting paper.

Mr. George Holmes (the Secretary): Professor Busley, the chairman of the German Institution of Naval Architects, is in this room. He is an authority on the subject, having written one of the most interesting papers that was ever published on the history and present position of submarine boats. He has asked me to say that he hopes to send his contribution to the discussion from Germany, because he does not sufficiently trust himself in the English language to make a speech to the members.

The following is the text of Professor Busley's letter:—

Captain Hovgaard's paper has the great merit of giving us, for the first time, a comprehensive and convincing mathematical investigation of the phenomena with which we have to deal in the movements of submarine boats. Just because it is so convincing, the fact stands out all the more clearly, that a submarine boat of 100 tons, moving at 6 knots, must dive by 36 ft. within a minute, if only *one* man walks forward. Mr. Sydney W. Barnaby has already referred to this point. I do not believe, however, as seems to be assumed, that such a rapid descent, and, conversely, also ascent, could be met by moving the rudder. My opinion is that any action of the rudder which is to stop those energetic motions could not but call forth a reaction, such that the descending boat will not return to the horizontal position, but will overshoot the limit, and begin to ascend. As a result, the completely immersed boat will never travel on a horizontal track, but will describe a wavy line approaching the horizontal. The few French reports which have been published grant this fact unreservedly, and hence the endeavours to convert the real *bateaux sous-marins* into the *submergés*. I may be permitted to point out that the first French submarine boat, the *Plongeur* of Admiral Bourgois and Naval Engineer Brun, built at Rochefort in 1863, had to be broken up again for this very want of longitudinal stability. It is simply inconceivable that a crew of two officers and nine men, to take the equipment of the most recent French boat *Narval*, would not stir at all during a half-hour's dive, such as Captain Hovgaard assumes in his concluding remarks, for a run of two or three miles under water. We should have to presume that the engine works with entirely automatic lubrication, without any attendance, or the slightest irregularity. I will not enter upon the question, whether the discharge of a torpedo

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would not affect the longitudinal stability to a much more marked degree than Captain Hovgaard appears to consider. He admits, moreover, without reserve that the introduction of a new torpedo into the tube must seriously disturb the equilibrium, and this will occur just at the moment when a strictly horizontal course is most desirable, namely in the face of the enemy, when the first torpedo has missed and a second better-aimed shot is to crown the attack. Since it is not to be expected that automatic devices will be designed and constructed which at once re-establish equilibrium, even after unintentional and inevitable movements on the part of the crew, we must necessarily reckon with an oscillatory motion of the boat. The strictly horizontal track will remain the desired, but rarely realised exception. But if the oscillatory motion is the rule, then the field of operation of the submarine boats, which is formed by the shore and the estuaries of ports especially to be protected, will be full of dangers to the boats themselves. There are not many estuaries without large banks of sand and silt; nor are there many shores with gentle and uniform slopes and without shallows. When a submarine boat runs into such a submarine upheaval—as happened, it is feared, to the *Plongeur* on the roadstead of La Palisse in 1864, and to Admiral Lord Charles Beresford and Campbell's *Nautilus* in 1887, when he got stuck in the Thames mud off Tilbury, and was not spared, finally, to Mr. Holland with his second boat in the New York sand in 1879—it must always be regarded as special good luck when the boat succeeds in clearing herself by her own efforts; without that, boat and crew are doomed. Nor is it, for these reasons, to be assumed that the idea of the American Lake, who wishes to let his *Argonaut* ride on wheels on the sea bottom, will prevail, although Captain Hovgaard supports it. Such a vessel would not get far on a muddy or silted sea shore, or where sea weeds abound, and those are the conditions met with on most shores, especially on the European coasts. But, supposing this trouble to be overcome. Submarine navigation still remains an unsolved problem, considering that the uselessness of the much-praised periscope has been established by the most recent French trials. There would be a last possibility, a kind of porpoise boat, diving up and down, fixing its course after having reconnoitred, and quickly disappearing again. But the modern boats, which are all submerged, except their conning-towers, still require two minutes for complete immersion, and there remain many difficulties to be overcome, before we shall be able to accomplish the instantaneous diving operations of which some inventors are dreaming. Not only on account of the deficient manœuvring power, however, but also on account of their want of longitudinal stability—and this defect would alone settle the question—the most recent submarines, the American *Holland* and the French *Morse* and *Narval*, have essentially been constructed as submersible boats which project from the water with their armoured conning-towers. It might require not a little courage to approach a modern armour-clad, with its accurate quick-firing guns, in a dome, about 2 ft. in diameter, at 6 or 8 knots in broad daylight, and at night the searchlights with their beams of 40,000 candles render such an attack a most doubtful piece of daring. The high-speed warship will, moreover, in a modern blockade be steaming out at sea, far away from the operation basis of the slow submersible craft. Again, the most recent gasolin engines, the heat, and the smell of oil, and the almost total absence of ventilation, make life on board a *submergé* unbearable, and the situation becomes dangerous when the vessel is surprised by bad weather. Both Mr. Hovgaard and Mr. Holland have, therefore, on previous occasions proposed to give the submersible boats a little freeboard, with the object of rendering the indispensable ventilation possible and of steadying the boat against the sea. If we adopt that, we lose the boasted invulnerability to hostile projectiles, and we may ask whether it would not be advisable to accept an ample freeboard. We could then pass to real seagoing ships, steaming at

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30 or 32 knots, instead of 6 or 8 knots, and the transition to the modern torpedo-boat destroyer would be complete.

Commander G. W. Hovgaard, Royal Danish Navy (Member): My Lord and Gentlemen, I feel greatly honoured by the kind way in which this paper has been received by the Institution, and I give my best thanks to the distinguished members who have been kind enough to take part in the discussion. Admiral Sir Nathaniel Bowden-Smith remarked that the submarine boats only go a very short distance under water. As far as I know, no submarine boats have made continuous runs of more than two, or perhaps three, miles under water. I also think that this limits the utility of such boats very much, though it is for many purposes sufficient. During attack, I think a submarine boat should be used in the following way. She should dive down for short distances, seeing her way perfectly clear before diving, and should not risk running under big battleships like the Italian boat did, but go on a certain determined course for a certain pre-determined distance with so many revolutions. She should then emerge and take her bearings, not being on the surface more than might be necessary to get a view of the enemy. She would then dive down again for another run, and so go on. When the boat is down, of course it is worse than in a thick fog, and there is no possibility of seeing anything. This method certainly limits the utility of submarine boats; but, notwithstanding, such vessels must have a certain value for harbour defence, and for coast defence within narrow limits. I do not imagine that submarine boats could generally be sent out to sea for great distances. There are two things which prevent that, namely, the endurance of the crew, which puts very narrow limits to the use of the submarine boat, and the endurance of the storage battery, or whatever source of energy is used for under-water propulsion. This must necessarily be very limited. I do not think, therefore, that these boats could be used for anything but strictly harbour and coast defence; for instance, against blockading squadrons, which may thus be kept at somewhat greater distances from the port. In landlocked waters, as in the Sound in Denmark, or in a great number of ports, as, for instance, in many of those of the English Colonies, where it may not always be possible to attack great squadrons of battleships, the submarine boats would probably be found useful. It would be well, I think, thus to have a cheap means of defence, which would always have a certain value in keeping off blockading squadrons of battleships from bombarding the towns. As regards the sliding along the bottom, that has actually been done; it is not a new idea. It has been done by *Le Plongeur*, the French submarine boat which was built about 1860. She slid along the bottom without any apparent difficulty. The "Lake" boat in America rolls along the bottom on wheels with a slight surplus weight of 200 lbs., and has done a lot of work in that way. It is claimed to have rolled for about 1,000 miles under water in Chesapeake Bay and other places without the slightest difficulty. The surplus weight can be regulated by pumping in and out water, so that the pressure of the wheels, or of the false keels, on the sea bottom can be regulated to any amount which is thought desirable. I do not see that there could be any great difficulty, provided only the bottom were tolerably even, and did not contain rocks or large stones.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, I feel sure you will all join with me in a hearty vote of thanks to Captain Hovgaard for his most interesting paper.

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# ON A FORM OF DOUBLE-TUBE BOILER.

By John Irving, Esq., Member.

[Read at the Spring Meetings of the Forty-second Session of the Institution of Naval Architects, March 29, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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Within recent years many papers dealing with water-tube boilers have been read before this Institution, but I now propose to describe the construction and performance of a boiler, differing very much from any hitherto brought under its notice, so far as I am aware. I trust that I may be acquitted of any intention of opportuneness in bringing forward a paper on boilers at this time. The fact is that this paper was both written and arranged for some time before the water-tube boiler question had reached the acute stage at which it has now arrived. About six years ago Mr. K. D. Noble and myself began to study the water-tube boiler very closely, with a view of trying whether we could not design a boiler suitable for the Mercantile Marine. A boiler that would combine the advantages to be gained from a use of higher pressures than are possible in a cylindrical boiler, and the economy of space in the ship, with simplicity of construction, and an accessibility of parts that would recommend it to more general adaptation than the forms of water-tube boilers then in use.

At the outset we determined to avoid all steel castings, bent tubes, complicated fastenings or difficult flangings.

When the idea of having the tubes double with a thin annular body of water between them, heated on both sides, first suggested itself to us, we at once saw the importance of determining the question of the possibility of overheating the outer tubes. We do not claim to have been the first or the only designers to use double tubes, but we hold that our arrangement of them is such as to give freer circulation than in any hitherto devised. For experiment a small furnace was built, in which we set two tubes 4 in. diameter outside, and $\frac{3}{16}$ in. thick, having a 3 in. tube passing through the centre of each, leaving an annular space of $\frac{5}{16}$ in. The arrangement of the whole apparatus is shown on Drawing No. I., Plate XLVIII.

Having made arrangements for feeding and firing the apparatus, we deliberately set to work to burn the outer tube if possible. With a strong coke fire, and a blast taken from the works' plate furnace blower, and equal to between 11 in. and 12 in. of

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water, we found the circulation rapid and complete. While the heat of the furnace was so great as to melt the fire bricks of which it was composed, the outer tube showed no sign of overheating.

Some time after this, when a new boiler was required for Messrs. Wm. Denny & Bros.' joiners' shop engine, electric lighting, and other purposes, it was agreed to give our boiler a trial.

It is with great pleasure that we take this opportunity of acknowledging our great indebtedness to Messrs. Denny, who have consistently encouraged the development of the invention, and generously borne all the expense and responsibility.

We at once set to work, and designed the boiler shown on Drawing No. II., Plate XLVIII. As this boiler was always regarded as an experimental one, its details were in some parts more elaborate and capable of variation than would be the case in one constructed on purely commercial lines. The outer tubes are 4 in. outside diameter, and ¼ in. thick. They are swelled at one end, and screwed into both inner tube plates, and secured with jam nuts exactly like the ordinary stay tubes in a cylindrical boiler. The inner tubes are 2¾ in. outside diameter, and pass through stuffing boxes in both outer tube plates. There are no inner tubes in the lowest row (see Drawing No. II., Plate XLVIII.). This was done first as a precaution: as the lowest tubes get the greatest heat, it was considered advisable that they should have the largest supply of water; and secondly, because the absence of an inner tube makes this most vital part more easy of inspection through the plug holes in the outer tube plates. The annular water space is three-eighths of an inch wide. The front water box is directly connected to the steam drum at its upper end. The lower end of this drum is formed into a box from which the down-comer tubes are led into the mud drum. This boiler being the first of its kind, it was thought advisable to provide a mud drum at the lowest part of the circulation, and so arranged that all the water would pass through it. The total area through the down-comer tubes was made specially large in order that circulation throughout the entire boiler should be as free as possible, and that, for experimental purposes, this area could be varied. The small Galloway tube connecting the top of the lower water box with the steam drum is for permitting the escape into the drum of any steam which might form in the top of lower water box. All the flanging and riveting are quite within the daily experience of boiler makers. The water boxes are stayed across with ordinary screwed fire-box stays; and, with the exception of the stuffing boxes for the inner tubes, which were an engineer's job, the entire construction was carried out by the foreman boiler maker with his ordinary shop hands. Pressure of other work delayed the completion of the boiler for some time, but it was finally finished, and passed the hydraulic test of 400 lb., in March, 1897.

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It was then decided that before being placed in position the boiler should be subjected to a series of the most exacting trials we could devise and carry out. For this purpose it was placed on the edge of the River Leven, removed as far as possible from all buildings. Tanks for measuring the feed-water were connected to the feed-pump, an excessively large fan was connected to the furnace, and we carried out the trials, the results of which will be found in Table I.

TABLE I.

FORCED DRAUGHT TRIALS OF DOUBLE-TUBE BOILER CONDUCTED DURING SUMMER OF 1897.

| Date of Trial. | July 7. | July 13. | July 14. | August 3. |
|---|---|---|---|---|
| Condition of boiler ... ... ... | Not lagged | Lagging on steam drum only. | | |
| Water evaporated per hour ... ... | 5,540 lbs. | 3,890 lbs. | 5,837 lbs. | 6,880 lbs. |
| Temperature of feed water ... ... | 59° | 62° | 60° | 65° |
| Mean steam pressure above atmosphere ... ... ... ... | 160 lbs. | 200 lbs. | 200 lbs. | 200 lbs. |
| Coal used ... ... ... ... | Scotch of fair quality. | | Welsh and Scotch mixed. | Welsh. |
| Coal burned per hour ... ... | 896 lbs. | 619 lbs. | 625 lbs. | 746 lbs. |
| Water evaporated per lb. of coal, actual... ... ... .. ... | 6·18 lbs. | 6·28 lbs. | 9·34 lbs. | 9·22 lbs. |
| Do. do. from and at 212° ... | 7·48 lbs. | 7·6 lbs. | 11·33 lbs. | 11·1 lbs. |
| Coal burned per sq. ft. of grate per hour ... ... ... ... ... | 71·75 lbs. | 47·5 lbs. | 48 lbs. | 57½ lbs. |
| Air pressure from fan, in inches of water... ... ... ... ... | 2·75 in. | 1·2 in. | 2·0 in. | 2·1 in. |
| Smoke-box temperature ... ... | 600° | 540° | 650° | About 650° (Melted lead but not zinc.) |
| Grate area ... ... ... ... | 13 sq. ft. | 13 sq. ft. | 13 sq. ft. | 13 sq. ft. |
| Total heating surface... ... ... | 820 sq. ft. | 820 sq. ft. | 820 sq. ft.* | 820 sq. ft. |

Ratio of heating surface to grate area, 63.
Spiral retarders were placed in all the inner tubes during these trials.

* At the end of this trial, steam was blown off, water run down, and an inner tube drawn and examined all in a few minutes. The tube was then replaced, and steam raised subsequently without any leaking or other trouble.



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The boiler was under steam for the greater part of each of the days noted, as well as on many other days when steam was got up for the purpose of making special observations. The feed-water generally in use was the town's gravitation supply; but special trials were made, feeding with the river water, to see if any deposit was left in the annular tube space. No grit, sand, or rust has ever been found there; the little we did discover was found mostly in the bottom of the lower water box and in the mud drum. The circulation in the tubes seems to be too rapid and complete for the deposit of any impurity. On one occasion we had ocular demonstration of the thoroughness of the circulation by getting up steam with the manhole door removed from the back end of the steam drum. As soon as the water began to get warm, it could be seen flowing in a full and constant stream from the upper to the lower end of the drum.

The boiler standing quite in the open air, and only partially lagged, was very far from being in the most favourable working condition. Much better results would have been achieved had it been placed in a closed stokehold, or formed one of a number of boilers arranged in battery. The results shown on the table speak for themselves, and at present need not be further alluded to. Although provision was made for the difference of expansion between the outer and inner tubes, we have never been able to detect any such difference: the fact that the hotter tube of the two is somewhat shorter than the other leads us to think that the difference is fairly compensated for in this way. The stuffing boxes form a convenient arrangement for withdrawing the inner tube at any time, so as to inspect the inside of the outer tube.

Steam was allowed to blow off from the stop valve in a steady stream. Its dryness we were very well able to observe, as it blew right over the surface of the River Leven. On a calm summer day no drop of water was ever seen to fall on the surface of the river. No priming has ever been observed, nor any sudden variations of pressure.

Two water-gauges were fitted, one to each water box. When steam was up, the one attached to the higher water box showed a lower level than the other, the reason of this being that, steam and water mixed having less weight than solid water, the head of water for circulation was always apparent.

The low temperature of the uptake gases is worthy of remark as showing how much heat was taken up by the water. Subsequent trials were made for the express purpose of verifying these results, and the boiler was at last passed as in every way fit for its intended duty.

Since its installation the boiler has been continually at work. At first it was worked on natural draught, but, as it was intended for forced draught, a small electric-driven fan was afterwards applied to the ashpit. The utmost this fan could give was

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about ½ in. of air pressure at its discharge, but even so small a pressure very much increased the efficiency of the boiler. It was now felt that the efficient working character of the boiler was tolerably well established, and it only remained to await with patience its emergence from a sufficiently protracted trial of endurance.

This, we consider, has now been accomplished. It has been in continuous use for over three years. Inner tubes here and there have been withdrawn from time to time for inspection, but no integral part of the boiler has ever required repair, much less renewal, and all the cleaning found necessary has been to clear the lower water box and mud drum through the hand holes provided for that purpose. This has not been done often, nor regularly; but about one-third of the water in the mud drum has been blown off once a week. No special supervision while working has been found necessary, one of the yard firemen having had charge of it all the time. No difficulty in feeding has ever occurred, an ordinary Worthington Duplex Pump, controlled by hand, having always performed this duty regularly and easily. Steam can be raised from cold water in about twenty minutes to the full working pressure. An automatic feeding arrangement and reducing valve were fitted at first, but found quite unnecessary. The coal used has been of the ordinary Scottish quality for steam raising, and, as with this quality of coal it was found better to have as large tube area as possible, the retarders were withdrawn. So far as we yet know, deposit, when there is any, sticks to the inner tube, not to the outer one; but, with regard to this, further experience with different kinds of water is necessary, as hitherto the town's gravitation supply has alone been used.

No zinc has been used in the boiler as yet. The same inner tube may be withdrawn and replaced any number of times. Outer tubes would be withdrawn only for renewal, which has not yet been found necessary.

The experience thus gained has suggested to us several alterations which are embodied in the amended design, Drawing No. IV., Plate XLVIII. The tubes and tube fastenings remain pretty much as they are, but the whole down-comer arrangement has been dispensed with, together with the mud drum. Each water box is connected to the steam drum direct. In the heart of the tube space a large combustion chamber has been formed, from which the gases are led away through a circular opening in the lower water box, to the lower ends of the inner tubes. A corresponding opening in the upper water box gives free access from the smoke box to the whole interior of the boiler. The construction of the furnace seemed to call for much attention, especially in the manner of admitting air to all portions of the grate. In boilers like this, where the fire bars are arranged in one large rectangular area, provision must be made for supplying air all round, both above and below the grate. As was stated at the

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outset, this boiler is intended for ordinary mercantile work; and, while it may compare unfavourably, as regards weight, with water-tube boilers in torpedo boats and other special light craft, it still remains very much lighter and more compact than the ordinary cylindrical boiler, even when the latter is worked under forced draught.

The following Table shows the relative weights, surfaces, size of stokehole, and other particulars of the ordinary cylindrical boiler, the double-tube boiler, and several of the better known water-tube boilers, as applied to produce equal powers in the same vessel.

TABLE II.

COMPARISON OF DOUBLE TUBE WITH CYLINDRICAL AND OTHER BOILERS, ALL ARRANGED AS FOR THE SAME POWER IN THE SAME STEAMER.

| Type of Boiler. | Cylindrical. | Double Tube. | Babcock and Wilcox. | Belleville. |
|---|---|---|---|---|
| Number of boilers ... ... ... | 2 | 6 | 4 | 10 |
| Height or diameter ... ... ... | 14 ft. 6 in. | 13 ft. 4 in. | 13 ft. 0 in. | 13 ft. 0 in. |
| Width ... ... ... ... ... | ..... | 8 „ 3 „ | 13 „ 0 „ | 8 „ 6 „ |
| Length... ... ... ... ... | 20 ft. 8 in. | 11 „ 6 „ | 13 „ 0 „ | 8 „ 0 „ |
| Heating surface ... ... ... | 9,133 sq. ft. | 12,060 sq. ft. | 12,840 sq. ft. | 13,380 sq. ft. |
| Fire-grate area ... ... ... | 300 „ | 234 „ | 286 „ | 479 „ |
| Working pressure, lbs. per sq. in. ... | 160 | 250 | 250 | 300 |
| Length of bars ... ... ... | 6 ft. 0 in. | 6 ft. 0 in. | 6 ft. 6 in. | 6 ft. 3 in. |
| Boiler-room area at floor ... ... | 1,065 sq. ft. | 1,008 sq. ft. | 1,035 sq. ft. | 1,334 sq. ft. |
| „ „ main deck ... | 384 „ | 276 „ | 332 „ | 468 „ |
| Heating surface per sq. ft. of boiler-room area ... ... ... ... | 8·54 „ | 11·3 „ | 12·4 „ | 10 „ |
| Total weight—boilers, uptake, and funnel ... ... ... ... | 174 tons | 156 tons | 130 tons | 186 tons |
| Total weight—water... ... ... | 84 „ | 12 „ | 17 „ | 14·5 „ |
| Total of boilers and water ... ... | 258 „ | 168 „ | 147 „ | 200·5 „ |
| Air pressure ... ... ... ... | 1 in. | 1½ in. | ½ in. | Nil. |

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## DISCUSSION.

Mr. A. F. Yarrow (Vice-President): My Lord and Gentlemen, I am sure we are all very much obliged to Mr. Irving for having given us this interesting paper, and having described a very ingenious type of boiler. We are also thankful to him for having brought forward a paper on water-tube boilers, a subject just now attracting so much attention. There are only one or two points to which I will refer. The author speaks of the low temperature of the up-take gases as showing how much heat is taken up by the water. I am sorry to disagree with him, unless he couples this statement with the composition of the gases. It is often erroneously considered that the low temperature of the funnel gases is an indication of the economy of the boiler, but if an excess of air is passed through the furnace and not burnt, it will be admitted that a very uneconomical process is followed. Any air which is simply heated up and not adding to the combustion wastefully absorbs heat, and at the same time lowers the temperature of the funnel gases. For these reasons the temperature alone, without an analysis of the gases, is not an indication of the economical absorption of the heat. It is stated as an interesting fact "that, although provision was made for the difference of the expansion between the outer and inner tubes, we have never been able to detect any." I think this is due to the fact that the chief resistance encountered by the heat (as it passes from the flame and the heated gases to the water) is due to absorption by the metal of the tube. There is very little resistance in the passage of the heat through the metal itself, and very little in passing it from the tube to the water. As a matter of fact, if the tubes are clean and the circulation good, I think we may say for all practical purposes (although at first sight it may not appear so, but from experiments we know it) that the temperature of the tube is really the same as the temperature of the water. It follows that both the inner and outer tubes would be the same temperature for all practical purposes, and any difference that may exist is not sufficient to exceed what the elasticity of the metal will be able to provide for. A very simple illustration of what I say may be given. We have all been brought up from our infancy to know that if one takes a tin saucepan that is soldered with a solder melting at a low temperature, so long as that saucepan is full of water one may put it on any fire, however hot, and the saucepan will be quite safe from destruction. That shows that the temperature of the metal has not been raised much, otherwise the solder would have melted. I think this important in considering boiler construction. To provide for variations in expansion in some forms of water-tube boilers, where the circulation is very active, is to provide for a difficulty which does not exist. Provided the surfaces are clear and the circulation rapid, I do not think we need trouble very much about this consideration as affecting the design of the boiler. I see the author says: "An automatic feed arrangement was fitted at first, but it was found quite unnecessary." I may mention that a very large number of cruisers and ironclads abroad have been fitted with Yarrow boilers, some of very large size. I think altogether there are twenty-six large vessels. At first automatic feeds were provided for those boilers, but of late years these have been abandoned. I think it is a complication that under most circumstances is really unnecessary, and I am glad the author confirms my views on that point. I should like to know, after the experience he has had, whether the author would use stuffing boxes, if he were to make other similar boilers to the one he describes, or whether his experience has indicated to him that he might dispense with them altogether. In the Table there is a comparison of weight: the cylindrical boiler was 174 tons, and the Belleville boiler 186 tons. I was under the impression that the Belleville boiler was lighter than

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the cylindrical type. I think the stokehold weights work out with the Belleville boiler as 24 H.P. to the ton, and with the cylindrical boiler about 18 H.P. to the ton, so that I do not understand how the author has arrived at the statement showing the Belleville boiler the heavier of the two.

Mr. J. H. McIlwaine (Member): My Lord and Gentlemen, I have in the first place to thank the author for this most interesting paper. So far as I know, the main idea of this boiler is entirely novel. The author offers us two designs: one he has tested, and another he has not tested. I presume he would argue that, because design No. 1 did well, therefore, the boiler represented by the drawings in Fig. 4 would also perform in the same way. I think that is an assumption that will hardly be carried out in practice. I will confine myself to the consideration of design No. 2, which has not been tried. In general terms, I think that the circulation of the heated gases would not be satisfactory, and that the same heat would not be abstracted with that construction of boiler as with the ordinary marine boiler, or as with Mr. Irving's No. 1 design. I think also that the circulation of the water would not be satisfactory. It is very probable that trouble would be caused by rushes of combined steam and water leading to priming. With these preliminary and general remarks I would ask you to follow a few criticisms of the paper, taking the various points in the order in which they are printed here. Mr. Yarrow forestalled something which I had to say about economy, and I need not repeat what Mr. Yarrow said, but merely say that I agree with him. I think Messrs. Denny's great kindness in trying this boiler is a compliment not only to the inventor, but a benefit to us all. We know that Messrs. Denny are always to the front when anything new is to be tried. The author says: "The outer tubes are 4 in. outside diameter, and ¼ in. thick. They are swelled at one end and screwed into both inner tube plates, and secured with jam nuts, exactly like the ordinary stay tubes in a cylindrical boiler." We all know that stay tubes, from one cause and another, give trouble. In this boiler we have all the outer tubes, stay tubes, and I believe we may look for a great deal of trouble. In the next line we come to the words "stuffing boxes." That suggests to my mind trouble of all sorts. It is putting the engineer's work into the fire, and whenever that is done one may expect trouble. I do not think stuffing boxes are a proper boilermaker's job. I have not time to state all the reasons for so thinking, but I believe any of those who have experience of what heat will do in similar cases will say that the boiler would be more satisfactory without stuffing boxes. Then, further down, it is said:—"The total area through the down-comer tubes was made specially large in order that circulation throughout the entire boiler should be as free as possible, and that, for experimental purposes, this area could be varied." That brings up the point that, in his second design, Mr. Irving has done away entirely with, I think, one of the principal and best features of the boiler, a means of good circulation. In another passage it is stated that "All the flanging and riveting are quite within the daily experience of boilermakers." So they are, but they are about as difficult a sort of flanging and of riveting as one can get; and the author does not mention caulking. I would not undertake to say, but I think some parts of the boilers would not get caulked on both sides, and we know that in modern practice it is most essential they should be. The next thing Mr. Irving says is that the water boxes are stayed across. Here he introduces, in about the worst position in which they could be placed, things all dread, even in a good position. Any person who has had experience of a boiler when it begins to pass middle life, and remembers the trouble there is in renewing these screwed stays, and who then sees by the drawing how many are in a bad position, will agree with me in concluding that trouble will be in store for the man taking charge of this boiler. Table I. simply deals with forced-draught trials, and I propose to pass that

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by, because it does not refer to the boiler I am discussing, that is design No. 2. In his paper, the author says :—"On one occasion we had ocular demonstration of the thoroughness of the circulation by getting up steam with the manhole door removed from the back end of the steam drum. As soon as the water began to get warm, it could be seen flowing in a full and constant stream from the upper to the lower end of the drum." That is exactly what we would expect, and exactly what we want, yet Mr. Irving, for some purpose, has done away with that in No. 2 design. We shall have the reason, I suppose, later on, but I think that is a very grave departure from the usefulness of the boiler. He says, again :—"The stuffing boxes form a convenient arrangement for withdrawing the inner tube at any time, so as to inspect the inside of the outer tube." It may be convenient, but I do not think it is a good, or a workmanlike, one. Then, referring to No. 1 design, the author says :—"All the cleaning found necessary has been to clear the lower water box and mud drum through the hand holes provided for that purpose. This has not been done often, nor regularly; but about one-third of the water in the mud drum has been blown off once a week." How is the same effect to be got in a boiler made to No. 2 design? How will the mud and accumulated dirt be removed? The author says at the end of that paragraph: "So far as we yet know, deposit, when there is any, sticks to the inner tube, not to the outer one"; and then there occurs a sentence with which I thoroughly agree—"but, with regard to this, further experience with different kinds of water is necessary." I think the experience needed is seagoing experience. Then in the next paragraph there is this :—"The same inner tubes may be withdrawn and replaced any number of times." I think that is sanguine. Then the next paragraph is just a reiteration of what I have referred to before. "The tube and tube fastenings remain pretty much as they are"—that is, referring to the construction of the boiler, it remains as it is—"but the whole downcomer arrangement has been dispensed with, together with the mud drum." I have to say again, I think that does away with the usefulness of the boiler to a very great extent. I merely quote this to show that I am not mistaking Mr. Irving's views. He says :—"As was stated at the outset, this boiler is intended for ordinary mercantile work." That I take to be tramp and ocean steamers, and the like. In Table II. Mr. Irving makes a very favourable comparison with his own boiler as regards dimensions and in several other respects. Is it possible to make this comparison? To do so you must cut the heart out of the boiler, and cut away the back end of the drum. If he has done that, I think he has made a mistake. I believe it would be better to take a little more room and have the boiler which the author designed first, and which I still consider better than the one he offers in No. 2 design. I would say that up to the present I do not think Mr. Irving has produced a successful rival to the Scotch boiler, but there is no doubt he has brought forward a boiler that may lead, in his hands, to further development and improved construction, and be a very useful boiler. There is one other point to which I would like to refer. In what is called a steam drum, in No. 2 design, the water-line would, I presume, be parallel to the base line of the boiler. In that case there would be very little water at the front end, and a great deal of water at the back end of this drum. This I consider would be a serious aid to priming, on account of the very limited area of the water surface. If the steam drum were level, the area would be a parallelogram half-way up the drum, equal to the length by the breadth of the drum. But in this case there is much less water surface. Any person who has had any experience with boilers will say that reduced area for disengaging steam from generating tubes always induces priming.

Mr. John Watt (Associate): My Lord and Gentlemen, I thank the author very much for the kind manner in which he has given us an invitation to discuss his paper. This subject is a very old

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and familiar one to me, from a great number of experiments. The boiler shown here, with the exception of the inner tubes, and, of course, the downcomers, is almost exactly the same as I had at work some twenty-five years ago, and, in fact, there are some boilers (now twenty years old) at work which are very similar to this. The author has just found out some of the little difficulties that I experienced, and he has improved upon them in his last design. This, with the exception of the inner tubes again, is almost identical with the last patents I took out, some ten years ago. Now the question is this: What advantage is obtained by an inner tube? The author has not stated this exactly. He gets certainly a little more heating surface, but I can, in my design, raise the boiler a little, and put in more tubes and heating surface. A boiler having one tube within another, with only $\frac{3}{8}$ in. of a space for water, will not work out in practice. The larger the body of water over the fire the greater the success of the boiler. Such being the case, how much heat does the author expect to get after the products of combustion have passed the outer tubes? The products of combustion may enter the inner tubes at about 600° or 700° Fahr., but the tubes will not take up much of that heat. It is somewhat similar to a series of hot water pipes. The difference between the outside air and these hot water pipes may be about 100° to 150°. If a tin of water be placed on the top of those pipes, and one tries to evaporate that water, most of it will remain unevaporated. It is the same with this design, because the spent gases are insufficient to evaporate much water. With a very high pressure, and, consequently, a high temperature, instead of the gases heating or evaporating the water, the water would be heating the gases. I see, too, that in one of the drawings there is a guard placed before the tube ends to prevent burning. Now, the author need not be afraid in the least of this taking place, because in locomotives there is a tenfold temperature there, and the ends of the tubes are not burnt. Therefore the guards might be dispensed with. Speaking from a practical point of view, I cannot see that the author will get much value out of the inner tube. In the experiments he has made the water has been from the River Leven. I believe that water is a surface water, and contains no lime. Now if this boiler were tried with water containing lime, such as sea water, there would be a deposit, to a certain extent, both on the inner and in the outer tubes, and the result would be that the boiler would burn. So far as I can see, this is a departure in the wrong direction. We want more water space over the fires to absorb the impinging heat, and we also want simplification. This would be a difficult boiler to clean, for all these inner tubes have to be taken out and replaced, whereas, in the ordinary way, the boilers I have mentioned can be cleaned in a few hours in the following way:—Steam is blown off, the top row of doors is removed, a sponge is put through the tubes and they are thus cleaned. Then the doors of the next row are taken off, the top ones replaced, and so on, right down to the bottom, sponging the tubes out. The whole boiler is thus cleaned in three or four hours.

Mr. Leslie Robertson (Member): My Lord and Gentlemen, the paper we have before us is in many respects interesting, and anybody who brings forward a paper on water-tube boilers at this moment is doing a service to the Institution. Before we pass on to the actual points of the paper, I should like to make a remark about what the second speaker said, namely that the subject-matter of this paper was novel to him. Mr. Irving said quite rightly that the matter was not a novel one, and that this boiler had been used before. I thought that it would be of interest to the Institution, and therefore I asked M. Bertin, the Chief Constructor of the French Navy, to let me have particulars of the tests that were made on a precisely similar boiler in the French Navy, and as they may be of interest to the Institution, I propose, first, to say a word or two as to what happened there. A boiler of

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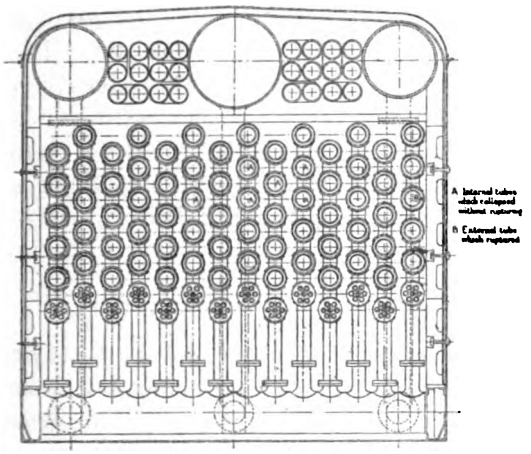
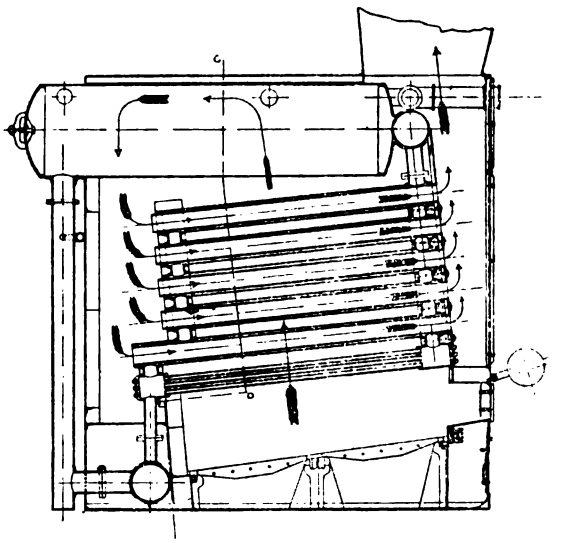
this description was fitted on a boat called the *Sarrazin*. The boiler is illustrated in Fig. 146, p. 281, of "Marine Boilers," by M. Bertin (John Murray), and is shown in the accompanying engravings. It was of precisely similar construction to Mr. Irving's boiler. There are the big upper tubes, the





slanting tubes with the concentric inner tube; the big downcomers; the mud drum at the bottom, and the furnace under the inclined tubes. The only radical difference between the two boilers is that Charles and Babillot, who fitted their boiler in 1894, used on the bottom row a whole series of small tubes. There were six small tubes instead of the one large single open tube shown on the bottom row of drawing No. 2, where Mr. Irving has wisely omitted his inner concentric tubes. The principal difference between Mr. Irving's bottom row and that of the Charles and Babillot boiler is that the bottom row in Mr. Irving's boiler next the furnace is fitted with one large tube instead of six small tubes, in both cases with no inner tube. The whole of the other boiler tubes are concentric tubes. The Charles and Babillot boiler was fitted on board the torpedo boat *Sarrazin*, and I am sorry to say the results obtained were disastrous both to life and to the boiler. What happened was this. One of the concentric outer tubes burst, and there was loss of life. But that was not all. The interesting point about it is that a very large number of the internal tubes, instead of keeping concentric collapsed, and if it had not been for the good material of which the tubes were composed, there would have been a good deal more trouble. These were the two principal difficulties that occurred. Of course, this boiler was working, not with Dumbarton water, which is extremely good, but on board a boat. I think, if only from a historical point of view, we have to thank M. Bertin for his kindness in placing this information at the disposal of the Institution; because, after all, we can often learn most from the mistakes of others. I will now pass to one or two other points. Mr. Irving has a proportion of heating surface to grate surface of 63, and he fits retarders. I do not think we can take a test with retarders as a fair test of the boiler under working conditions, because he admits afterwards that he takes out his retarders; and, further, the evaporation that we get of 7·48 lbs. of water per lb. of coal form and at 212° Fahr., although fair, is not a record by any means. The same remark, I think, holds as to the temperature of the funnels. They are pretty fair; but, without the composition of the gases, it is very difficult to know what is actually taking place.



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No doubt Mr. Irving will try to test his boiler with very bad water, calcareous water, not Dumbarton water; then he would learn where the deposit actually takes place. I think, myself, the circulation would be so rapid between the concentric tubes that the scouring action would have the effect of keeping them clean and free from deposit, except in those tubes where the evaporation and the scouring action would not be so strong. That Mr. Irving has been wise in keeping out the internal tube at the bottom, I think is quite clear, from the fact, which is no doubt known to most in this room, that over 22 per cent. of the total evaporation in the boiler is done in the bottom row of tubes next the fire; and, therefore, the upper rows are doing very much less than their fair proportion of the work. It is rather interesting to notice that Mr. Yarrow's experiments on length of tube and expansion exactly confirm the result arrived at by Mr. Irving. There is one point which may perhaps throw some light on the matter. I think the constructional details are often the true secret of the success, or otherwise, of the boiler. In the French boiler, which had such unhappy experiences, the internal tube was longer than the external tube, and not, as Mr. Irving has it, shorter.

Mr. Irving: I beg your pardon, my internal tube is longer.

Mr. Leslie Robertson: I thought it was the other way; I am sorry. The author is quite right in admitting air above the grate, and I think several of our boiler constructors have found out that they have made a mistake in not doing so. In the second design, which has had to bear the brunt of the criticism so far, I think that the bars are too close to the tubes. Mr. Irving would get much better results, if he would drop his furnace a good deal. It does not do to hurry the gases in amongst the tubes too soon, especially if they are working under forced draught, as this boiler is designed to do. In the Table, Mr. Irving assumes 1½ in. of draught for his boiler as against nothing for the Belleville boiler, and ½ in. on the Babcock and Wilcox boiler. If the author ever constructs the boiler shown in drawing No. 4, and if he adheres to the dimensions which he has shown at the back end of the bars, I think he will find that, although there is a secondary combustion chamber, the bars are too close to the bottom of the tubes, and he would get a much better result if he did not keep them quite so close. I should like to ask how Mr. Irving inserts his stays inside the water spaces, how he gets the rings in, how he keeps them in position, and whether he has any trouble in tightening them up. No doubt there is some method of getting over that difficulty. I should like to ask also what, in the second design, is his reason for putting the steam outlet at the back right up over the top of the drum. In drawing No. 4 there is a steam space shown right round in a ring. Does the author think any steam will be given off from that end of the boiler? Surely, if the circulation is correct, the steam should all come from the other end.

Mr. C. H. Wingfield (Member): My Lord and Gentlemen, I should like to ask the author how he gets his outer tubes in, and how he gets them out again for repair and replacement? I notice in the Table of tests that different kinds of coal were used. This makes it a little difficult to compare the results. In the natural order of things, as one forces a boiler more steam is generated; but the efficiency falls off, and more coal has to be burnt in proportion. When the author's boiler was under its best conditions for working, I see he used the worst coal; when it was working at a medium rate its performance would naturally fall off, but he has used rather better coal; when it is working the hardest of all, and the performance, other things being equal, should drop a good deal, he has used the best coal he could get. That makes it impossible to judge how the efficiency is really affected by forcing. In the Table of comparative weights the air pressure with his boiler is put rather high as

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compared with others—it is 1½ in. I suppose there is some justification for suggesting the boiler would stand that continuously? The tests recorded in the paper go up to that pressure for a short time, but during the three years' trial, on which I believe the author bases his faith in the boiler, it was run with a small fan, only capable of giving a maximum air pressure of half an inch. This was not a test which could give any indication of its behaviour under the harder conditions proposed. Mr. Robertson said that the French boiler, of which he exhibited a drawing, was, with one exception, exactly the same. I think that there is another exception. In the author's boiler a Galloway tube has been provided to allow any steam to escape which would otherwise have to pass through the generating tubes. In the French boiler that Galloway tube is not provided. In that respect I think the author's boiler is the better of the two. The device might possibly be advantageous, although I believe that the trouble with boilers of this type comes largely from accumulation of steam in the upper halves of the tubes in the lower rows. A question about the boiler staying has been asked. I should not be disposed to quarrel with the author for using stays, as they are quite satisfactory where properly designed—for instance, in ordinary Scotch boilers. I should like to point out, however, that in his first boiler the stays had very little to do. I believe the author did not use stuffing boxes in the boiler which he tried, but that the inner tubes were expanded into the outer plates, and the outer tubes were expanded into the inner plates. They would thus largely do the work of the stays, and, even if the stays were bad, they would not be very much strained where stuffing boxes are fitted. As now proposed, the stays would have all the work of holding the sides of the water box however. On referring to the trials on July 7 and July 16, it will be seen that in both instances the boiler was evaporating about the same amount of water per hour from and at 212° per foot of heating surface. In the first it was 8·17 lbs., while in the second it was 8·63 lbs. Other things being equal, one would expect to find nearly the same efficiency. The coal, however, was not the same, as already pointed out. In the trial of July 16 the evaporation is given as 11·33 lbs., while in the other trial, at the same rate per foot of heating surface, it was only 7·48 lbs. per lb. coal. That was partly due to the coal, but I think there was also another reason. The air pressure seems to have been high on the better trial. The ratio of the square root of the air pressure to the coal burnt per square foot of grate per hour is an approximate measure of the resistance which the air meets with in being forced through the boiler; it is, therefore, a rough indication of the thickness of the fire. If the damper was not touched, I should expect to find that the fire was thicker on July 16 than on July 7; that is, that this boiler did better with a thick fire than with a thin one. It is possibly, therefore, not a bad thing to use a thick fire in a boiler with a low firebox, but provided with a supplementary combustion chamber. In a boiler where there is ample room in the firebox for combustion, however, a thin fire always does the best with forced draught. Allusion was made (by Mr. Yarrow, I think) to the comparative weights given by the author for the cylindrical boiler and the Belleville boiler. The author has included the weights of the uptakes and funnels. There are ten uptakes in the Belleville and only two in the cylindrical boilers, and by including the weight of these the relative weights of the different types of boilers are effectively masked. It is therefore impossible to make a useful comparison. The following particulars of an installation of the new Thornycroft-Marshall boilers (which were lately illustrated in *Engineering*), of the same power as those in the author's Table, may be of interest. The height —measured to a flange some distance up the uptake—is 14 ft. 2 in.; the boiler proper is only 13 ft. 8 in. high. The width is 9 ft. 9 in., the length near the floor level being 8 ft. about. These compare favourably with the author's figures. The heating surface of the ten Thornycroft-Marshall

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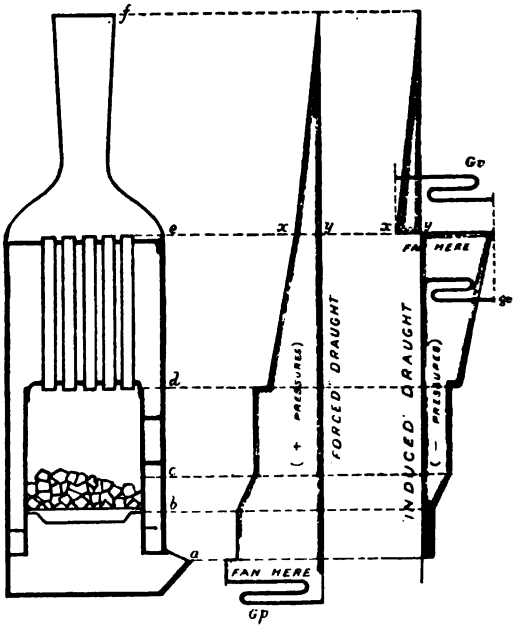
boilers is 12,000 square ft., the fire-grate area is 325 square ft., the working pressure is 220 lbs. per square in., and the length of bars 5 ft. The author does not say what power he has assumed, but I estimate that the boilers he has put down in his Table ought to evaporate about 100,000 lbs. of water per hour.

Mr. Irving: 5,000 H.P.

Mr. Wingfield: Yes; it is on that assumption I have based these figures. The total weight (dry) of the boilers and fittings, if of the Thornycroft-Marshall type, without funnels, is 142½ tons. The weight of funnels depends on the ship more than on the boilers; if 7¾ tons, the total weight (dry) would be 150 tons. The total weight of the water is 30 tons; the air pressure, or rather the vacuum, is 0·3 in. There is rather an interesting point about trials made under induced draught. Many of the advocates of induced draught say that with it you get much more steam, for a given water gauge, than by "forcing," and very fantastical reasons have been given for this. Other people have said that is not so, and that better results cannot be obtained, because a given difference of pressure on the two sides of the fire, whether produced by pulling or pushing, must result in burning exactly the same amount of coal per hour. It occurred to me lately that there is perhaps a little more in it than that. I will take the case of a vertical boiler as the simplest form. Supposing the boiler were working under a pressure of air in the stokehold, the pressure being shown by the gauge G $p$, there would be a loss of pressure at $a$ due to velocity of entry at the ashpit door, a further loss of head in overcoming the resistance of the fire, as indicated in the fall from $b$ to $c$, and succeeding losses at $d$, $e$, &c. The head $x$ $y$ represents that necessary to overcome the resistance of the chimney, the pressure at $f$ being that of the external air. The second curve shows the pressures if the same volume of air were moved per minute by a fan placed at the base of the funnel. The fall of pressure is here the same in amount at each point, but the curve commences at atmospheric pressure, and falls more and more below it until the fan is reached, above which the *plus* pressure, necessary to drive the air through the funnel, is, of course, the same as before. The total gauge, as shown by a manometer G $v$ would be the same as before, but the reading actually taken is represented by a manometer $g$ $v$, which is less than the true reading by an amount corresponding to the resistance of the funnel. The diagram shows how it is that a "vacuum" = $g$ $v$ is said to enable as much coal to be burnt as the greater pressure necessary with forced draught. The fact being that the fan has actually to produce precisely the same difference of pressure (= $G$ $v$ or $G$ $p$) in each case.




Mr. G. W. Manuel (Member): My Lord and Gentlemen, I have very little to say, because I think the particular points have been pretty fully discussed by the previous speakers; I would like, however, to make a few remarks as to comparison with regard to previous boilers of this type, as Mr. Irving's description reminds me very much of a boiler I saw in Marseilles ten years ago, the

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Langrafell and D'Allast type used in the Navy and Mercantile Marine of France. With the exception of the internal tubes, this boiler was very much the same as that now described, though simpler, on account of not having the inner and outer tubes. There were also solid stays that supported the outer flat end and the tube plates. In this case, of course, the small tube stays are intended for the outer plates. The difficulty they experienced in France arose through those solid stays and plates corroding, owing to the rush of water and steam from the tubes up towards the steam drum. The stays and plates deteriorated very rapidly. Judging from the drawing of Mr. Irving's boiler, it would be rather worse, because the small tubes, also coming through the water space, would cause rather more friction and wear on the stays. I take more interest in the comparisons here made with the mercantile circular marine boiler, because this boiler is stated to be used perhaps as for mercantile purposes, and I may have the pleasure of meeting it some day on board ship. But the results so far given can hardly enable me to form an opinion of what it will do when it is at sea. The water of Loch Lomond is water something like the water of Loch Katrine, it is very good for boilers, being almost distilled water and free from organic matter. Therefore a boiler fed with that water would last much longer than a marine boiler at sea. It is not mentioned whether this was a cylindrical boiler with forced draught.

Mr. Irving: There was one inch of air pressure.

Mr. Manuel: It is not stated whether it is Howden's forced draught boiler. We have heard much about the Scotch boiler and the tank boiler, but I should like to see work done by these modern tubulous boilers compared with what we call the most advanced circular marine boiler, that is Howden's forced draught boiler as used in the mail services. A comparison with the Belleville boiler is also made. The Belleville boiler is quite different from Mr. Irving's. The tubes of Belleville boilers are about 50 ft. long. In this case the tubes are shorter, 11 and 14 ft., and more resemble those of Mr. Yarrow's straight tube boiler. If Mr. Irving were to take some of the results of the Langrafell and D'Allast and Yarrow boiler, and compare them with his own, he would get comparisons a little closer to the object he has in view. The absence of simplicity in his boiler arises through using two tubes, and at each end of the tubes too many separate jointing pieces. The D'Allast boiler is just a tube put into the tube-plate in the ordinary manner. There is nothing else. These numbers of jointings are rather complicated. There is another point, the only baffle plate Mr. Irving appears to have is that on the top. Is that so?

Mr. Irving: Yes.

Mr. Manuel: If the tubes were longer and more baffle plates were used, the gases would be made to pass circuitously among the tubes, with, I think, better results. No. 4 drawing shows a combustion chamber. It is what all tubulous boiler designers have been trying to get lately. It has been found that the rapidity with which the gases pass vertically among the tubes has been such that they get up to the funnel unconsumed, and are not ignited till above the tubes. In some cases of hard firing they get to the top of funnel without being burnt. The great efficiency of any boiler arises through the ignition of the gases in the boiler flame spaces. I hope when the boiler is tried in a sea-going vessel the author will be able to give us further results.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G.): I am afraid I must ask you now to allow the discussion to be interrupted for a few minutes. I am called to another place, and I would ask Sir John Hay to take the chair.

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Admiral the Right Hon. Sir JOHN DALRYMPLE-HAY, Bart., K.C.B., D.C.L., F.R.S. (Vice-President): Gentlemen, as this is the last occasion we meet during this Session, I think you would wish to thank our President for his services before he has to leave the chair to go to his duties in what he calls another place, but which I will call the House of Lords, as I am not forbidden to say so here. As Lord Glasgow is called to go there, I think you would wish, at the conclusion of the Session, to offer him a vote of thanks. I would like to say this, that, when Lord Hopetoun left us, we were in great dismay. You all know how admirably Lord Hopetoun conducted our affairs for some years. However, among the many good pieces of advice which he gave us was this, that of all persons he knew there was no person who could take his place with such satisfaction to the Institution as Lord Glasgow. He had known him when he (Lord Hopetoun) was governing Victoria, and when Lord Glasgow was governing New Zealand, and he had observed the admirable manner in which Lord Glasgow conducted the matters connected with the government of that Colony. I myself, when I heard the House Committee had determined to approach Lord Glasgow with a view of asking him to take the chair of this Institution, also remembered that in the part of Scotland with which he and I both had something to do, he had for many years been Chairman of the County Council of Ayrshire, consisting of 170 members, the largest in Scotland, and almost in the kingdom; and I knew how admirably he had done his duty there. I also remember—he being quite a young fellow, as you see he is—when he was a young officer in the Navy, and when I was an old one, that he did his duty as a Naval officer entirely to the satisfaction of the profession before other duties called him to his native country. I think that we are exceedingly fortunate in obtaining his services, and I am sure that the Institution will feel that, new though he is to the work, he has done it thoroughly well during this Session. I would also say this: there is another great Association, the British Association, which meets in different parts of the country. There they have always, during the more than fifty years which they have existed, asked some person of eminence belonging to the town at which they are going to meet to take the chair during that annual meeting. We know how advantageous it is to us to have a person who is a man esteemed and respected in the neighbourhood to preside over us. We may well remember on the banks of the Tyne how satisfactory it was to have Lord Ravensworth with us as our President, and how much that enhanced the manner in which we were there received. We are now going to the banks of the Clyde, and I can assure you that the noble Earl, who is now presiding over us, is as much beloved on the Clyde, and in its neighbourhood, as Lord Ravensworth was on the Tyne, or as any person could be in any part of the country. I see my friend Mr. Scott below me, and I am sure he would confirm the fact. If you asked the Dennys, of Dumbarton, the Thompsons, of Clydebank, or the Napiers, or the Fairfield Company, or those of Glasgow University, Professor Elgar, Lord Kelvin, and Professor Biles, and others, they will say that there is no person who could preside over us at our Glasgow meetings with such satisfaction, both to us and to the great city which is going to receive us, and from which he takes his title, as my noble friend the Earl of Glasgow. I beg you, therefore, to give a hearty vote of thanks to the Earl of Glasgow for having presided over us this Session.

Mr. BENJAMIN MARTELL (Vice-President): Gentlemen, I feel it a very great honour, as an old Member of the Council, to be allowed to second this vote of thanks to our President. I am sure you will highly appreciate the remarks of Sir John Hay, and will have agreed with him in every word that he has spoken. I can endorse what he said with regard to the difficulties we had on the retirement of Lord Hopetoun, and the very great satisfaction that we experienced in having the Earl of Glasgow to succeed him as our President in this Institution. He has shown us, although he has not been long

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in that office, that he is a nobleman of great common-sense, of great fairness of mind, and of great courtesy of manner, and we appreciate him very much indeed as a successor to Lord Hopetoun, and to our previous Presidents in whom we had so much confidence. Therefore, gentlemen, I have a very great deal of pleasure in seconding this resolution of Sir John Hay's, and I am sure you will mark your appreciation of the wisdom which has been displayed by the Council in selecting a nobleman with so high attainments, and with capacity for high and responsible office such as the Earl of Glasgow has shown.

[The resolution having been put to the meeting was carried with acclamation.]

The President (the Right Hon. the Earl of Glasgow): Sir John Hay, Mr. Martell, and Gentlemen, I have to thank you most cordially from the bottom of my heart for the manner in which you have received the very kind words—the more than kind words of my old friend, Sir John Hay. However, I think you will agree with me that when he compliments anybody he lays it on pretty thick. At any rate, I have carried out the business of the meeting to the best of my ability on this my first appearance in the chair, and, if I have satisfied you, I shall be more than rewarded for any little trouble which it has given me. I shall continue to do my best to fill this chair as your President as long as you think fit to extend to me your countenance.

Sir John Hay then took the chair.

The Chairman: We will now continue the discussion on the boilers.

Mr. G. Halliday (Visitor): My Lord and Gentlemen, I agree with Mr. Robertson and Mr. Wingfield that this is hardly a new type of boiler; in fact, it has been tried by a number of people in France, Germany, and the United States. It will be noticed in that design that it consists of a water-tube boiler and also of a flue-tube boiler, and therefore partakes of the nature of a cylindrical boiler and a water-tube boiler. Now, I think it will be allowed that the cylindrical type of boiler can stand ever so much more forcing with regard to the hot gases than a water-tube boiler, and therefore in this design the gases begin to heat the tubes the wrong way. Since the inner tubes are like the flue tubes of the cylindrical boiler, the better way, I think, would be to take the gases through the inner tubes first. They can stand ever so much more forcing and greater heat, and therefore they could stand the gases in their hottest state going through them first. Another way would be for the back of the grate to be arranged so that the fire gases could be taken through a passage and split above the fire, allowing a part of the gases to go through the flue tubes and another part up the back into the combustion chamber, and then through the inner tubes. In that way there would be obtained a combined flue-tube boiler, similar to the cylindrical type, and also a water-tube boiler. It would be possible to force it, or at least to get a greater consumption of coal. The arrangement which I should like best would be this—of course there are two practical difficulties always—to shut off by a damper the gases from the water tubes, especially when steaming under cruising conditions, and to let the gases play against the tubes in the same way that they would against the upper surface of the furnace; but to allow them to pass through into the combustion chamber at the back, then through the flue tubes, and then up the uptake to the funnel. That would be a very good arrangement, and, if the boiler were used in that way, the difficulty present in all water-tube boilers would be overcome. The drawback to the water-tube boiler, especially with a large tube boiler, is that it only lives for six years. A smaller tube boiler of the express type lives, according to the latest I have seen about it, for 1,500 hours. Now, a water-tube boiler of any kind will never be successful

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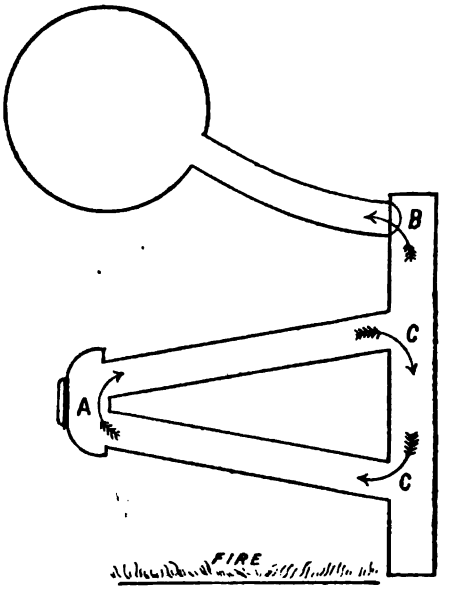
until its life can be made a little longer than that. Mr. List, in a speech which he made at the Civil Engineers', said that a cylindrical boiler in a liner would last easily, taking proper care of it, for fifteen years, and I have heard of them lasting twenty years; the tubes being quite good at the end. It would not be possible to get the tubes of the single water-tube boiler to last for anything like that time; in fact they are quite spent by the end of six years, and sometimes before that. Until we get a boiler for the Navy and the Mercantile Marine which will have a longer life than the present water-tube boiler, it can hardly be said that engineers have brought the question to a successful issue. I think, if this type of boiler were used as a flue-tube boiler, instead of a water-tube boiler, that is, if the gases were taken from the fire through to the back of the combustion chamber, it would give a combustion chamber such as water-tube boiler users always grumble about not getting, and also would give a boiler equivalent in nature to the cylindrical type of boiler as far as the heating surfaces are concerned. It would also give all the best features of the cylindrical type without its tremendous weight. The question that arises is, whether the space between the two tubes would stop up when salt leaked into the boiler. As to weights, I have seen a Thornycroft-Marshall boiler which appeared to me to be an excellent arrangement, and absolutely new. The tank, which was at the back of the tubes, was arranged as shown in the illustration (annexed). There was a joint at A, and the water circulates as shown by arrows above the fire, and goes back in the direction of arrow C. The novelty of this boiler appears to lie in this, that the circulation of the water never requires to go through the drum and make the steam so wet as it usually does in water-tube boilers. I think that is rather a new feature, and a very excellent one.




Mr. J. Macfarlane Gray (Member of Council): I do not like this boiler, I do not like the annular tube; but in one respect this boiler is better than the other water-tube boilers. Here the boiler is not divided into separate elements which can rob one another of their proper feed without the knowledge of the engineer. Nearly twenty-eight years ago I made an official report to the Board of Trade on the disastrous failure of the water-tube boilers of the steamship *Montana*. The substance of that report is published in *Engineering*, pages 366-368, October 31, 1873. The cause of failure is there pointed out in the following words:—"The mistake seemed, in the opinion of the officers of the Board of Trade, to hinge on the arrangement of feed. If, instead of having the different sections all open to each other at the bottom, they had been open to each other only at the second tube from the top, then, whatever might have been the irregularity in the evaporative capacities in the different sections, it would have been impossible to drive the water level below this communication, so long as a supply of feed was maintained." In the case of the Belleville boiler, what is meant here is, if the horizontal feed box of the Belleville boiler, now at the low ends of the sections and connecting them all, were raised to the level of the second tube from the top, and a good size feed pipe led separately from this box to the low end of each section, then, as long as the required head of feed level is maintained over this feed box, the water could not be driven from section to section through the feed connections, blindly, as it is at present in the

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Belleville and similar boilers. This arrangement of feed box and separate pipes perhaps would be impracticable; if so, the boiler without this arrangement ought to be also regarded as impracticable, and abandoned. It is astonishing that after the publication of the *Montana* report by one Department of the Government, another Department of it should repeat the same error in sixty battleships, entirely disregarding the warning given, the experimental result of an honest expenditure of £60,000 by an enterprising steamship company.

Mr. Archibald Denny (Member of Council): Mr. Chairman and Gentlemen, I am not a boiler expert, and therefore, perhaps, I can speak more dispassionately than some of the others who have addressed you. I am sure we all thank Mr. Irving for bringing this before us. It is quite refreshing every year to have a new discussion on water-tube boilers; it seems an endless subject. With regard to this boiler, I am perhaps responsible directly for having had one built. I was not very particular what water-tube boiler my firm built so long as they built one, and I thought our own friends should have a little encouragement, and, again, we needed a boiler in the yard at that time. Before it was sent into the yard it was tested, and everything was done that could be done to destroy it, finishing up, I believe, by feeding it with a mixture of two-thirds mud and one-third water; I believe, also, salt water was tried. Mr. Manuel stated that Loch Lomond water did not afford a very good test for a boiler, so far as endurance was concerned; and he is quite right. I have known boilers on Loch Lomond nineteen and twenty and twenty-five years old, and they had a bloom on the iron just as when they were built; but we do not use Loch Lomond water in Dumbarton, we use Loch Humphrey water, in which there is a fair amount of lime. What I should like to say is this: that, while I agree that the three years' test made with this boiler proves it to be an excellent land boiler, yet it in no way proves that it will be an excellent sea boiler. It has not passed that crucial test which should be applied to all water-tube boilers, of a leaky condenser admitting salt water into it. I think Mr. Irving's one desire is to have some adventurous shipowner try the boiler, so that this Society may benefit by that experience. I would just like to repeat that, so far as my own firm is concerned, the use of the boiler has given us the utmost satisfaction. It has had practically no repairs, and it has this advantage, that it can be very much forced. If more steam is needed, and there is a heavy demand for electric light in winter, it is simply necessary to put on more forced draught. As a land boiler I believe it is good, and as soon as an adventurous shipowner comes along, Mr. Manuel might perhaps induce his firm to try it; then we shall get additional experience.

Mr. John Irving (Member): Mr. Chairman and Gentlemen, if it was my main object in writing the paper to get this boiler well discussed, I think I have every reason to be at least content. With regard to the forced trials which appear on Table I. They were not intended to be so complete in detailed results as is only possible when experiments of this kind are conducted with great elaboration. All we wanted to know was whether our boiler would, or would not, make steam and keep tight at 200 lbs. We were satisfied when it evaporated the amount of water which we have shown without leaking or priming. If air was then passing from the furnace right through the funnel, the boiler will give even better results when that defect is remedied, as it easily can be; but we do not think it existed. I quite agree with Mr. Yarrow as to the question of expansion and automatic feed. I am glad to hear that he has been able to dispense with automatic feed apparatus, because everything of that description which can be eliminated from the boiler is a relief to the mind of the man who has charge of it. Then the stuffing boxes seem to require some explanation. I do not know



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whether my drawing shows them quite so clearly as I would like, but when there are two tube plates with a large tube finishing in the inner tube plate (as shown in drawing No. 2, page 201), the hole in the outer tube plate must be big enough to let the large tube pass. But as this hole is only partially closed by the inner tube, which is smaller in diameter than the hole, it is necessary to put something in to fill up the space. It was the most natural thing, as it seems to us, to make a stuffing box, and so provide at the same time for the expansion and easy withdrawal of the inner tube. A great deal of apprehension was exhibited as to what trouble might arise with these stuffing boxes, but surely it is not beyond the power of engineers to make stuffing boxes that would be light. Some doubt was expressed of the value of the inner tubes as effective heating surface. I may state that during the forced trials the retarders were red hot for a considerable length at the lower end, showing that the inner tubes were doing a fair share of the whole work. With regard to the details of Table II. the weight of the Belleville boiler was taken from data found in the Transactions of the Institution, and from the published descriptions of the *Powerful* and *Terrible*. If there is any authoritative correction to be made in this Table I shall only be too glad to incorporate it in the paper. As for blowing off, Mr. MacIlwaine, I think, expressed some dissatisfaction, especially with the amended design of No. 4. It is quite easy to make the back water box deep enough to provide a receptacle where deposits would accumulate and be blown off; in fact, the lower part is a good deal deeper than the front one for that reason, although the small scale of the drawing does not make that so clear as we had intended. A good deal of the criticism has been mutually destructive. I was very glad to hear from one speaker that a boiler of a somewhat similar design lasted for twenty years. That is a very different statement from that of another speaker, who prophesied three years, I think, or somewhere in that neighbourhood. It was said, on the one hand, that all screwed stays were bad, while another speaker said they did very well in locomotive and other boilers. The tube I have lying here has never been touched since the boiler was built, and I have been assured by those who have experience in post-mortem examinations of boilers that there is ten years of life in it yet.

Sir Nathaniel Bowden-Smith: When was the boiler built?

Mr. Irving: It has been working three and a half years, and the tube has been in position all the time.

Mr. Macfarlane Gray: Night and day, or day only?

Mr. Irving: You will pardon me; I think you mean, was the boiler working night and day?

Mr. Macfarlane Gray: Yes.

Mr. Irving: No; the fire was lit, and steam raised every day. With regard to the unlucky French boiler, the patent of Messrs. Charles and Babillot, our attention was drawn to it at an early stage, and an examination of their design showed such great differences in construction and arrangement as fully convinced us that there was no possibility of the defects it had shown ever arising in our case. A great many friends, whose opinions we received with the utmost respect, were unanimous almost in assuring us that this annular space would be a source of trouble, and that the steam would not get out of it; that it would froth there and that the tube would burn, and so on. I am glad to say that there has never been the slightest sign of anything of the kind, not even under the severest trials to which we have been able to subject it. It seemed to us that if one can be sure the water is getting down, and that the steam and water are rising up freely—in other words, if there is assurance

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of circulation—there need be no fear with regard to annular space.* In fact, an examination of the tube lying here will show that it is absolutely scoured at the bottom by the rush of water. It has been remarked that the retarders were first put in, and then taken out. It is quite true that when we were using Welsh coal, and were trying to get the highest evaporation out of our boiler, we had retarders in; but at that time we had 2½ in. of air pressure on. Afterwards the retarders were removed when the boiler was in daily work, and the demands upon it were not so great; there being only half an inch air pressure, whilst burning an inferior and smoky kind of coal. The stuffing box affords really very great facility for examining the boiler, because the tubes can be drawn and wiped out and put in again very quickly. One gentleman asked me to explain how the stuffing boxes are put in. The nut of the stay tube is a round nut with two slots on the face of it; not a hexagon. It is let down from the steam drum before the inner tubes are put in, and, with a hand put through the hole in the outer plate, a man can easily thread the nut on. It is then screwed up tight with a key suitable for the purpose. In answer to Mr. Manuel, I would say that the cylindrical boiler, with which we have compared our boiler, is an actual example at present working, and is in a closed stokehold. It is not a Howden boiler. Mr. Halliday thought the design would be better if the flow of the gases were reversed. If he can construct a boiler like that, I, for one, should be very glad indeed to see it, and, if he reads a paper on it here, I shall have very great pleasure in coming to criticise it. I thank you very much for the cordial acceptance which my paper has had at your hands, and for the very full discussion that has taken place upon it.

The Chairman (Admiral the Right Hon. Sir John Dalrymple-Hay, Bart., K.C B., D.C.L., F.R.S., Vice-President): I am sure the meeting will desire me to express its thanks to Mr. Irving for the interesting paper he has submitted to us, and to him and to Mr. Noble for the careful experiments they have made; also to Messrs. Denny for giving them the facilities necessary to bring the boiler to such a stage. The thanks of the Institution, I am sure, will be accepted by Mr. Irving, and conveyed to Mr. Noble for the very valuable information which they have given us.

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* In connection with this point, it is worthy of note that the Dürr and Niclausse boilers at present suggested for trial in the Royal Navy have double tubes, which are said to work very well, and in both of these the annular space is closed at one end, while in our design it is through-going.

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## CONCLUDING PROCEEDINGS.

Mr. J. Macfarlane Gray (Member of Council): Gentlemen, our noble President has left us and gone to his Hall of State. "This wide and universal theatre presents more pageants than the scene wherein we play in." But our little play comes to an end too, and we have now to end our meeting, to leave this painted hall, a place which is very agreeable to me, and brings back to me many pleasant memories since the year 1861—forty years ago—that I came first to our meetings here, and I have been a very frequent attendant since. It well becomes me, then, to be the mouthpiece of the Institution, to ask you to give a hearty vote of thanks to the public-spirited Society of Arts, who have given us for all that time the use of this beautiful room, and who have given us so many facilities to enable us to feel at home—just as comfortable as if we were in our own house. I ask you, therefore, to give a hearty vote of thanks to the Society of Arts for their kindness to this, a kindred Institution to their own.

Mr. Sydney Barnaby (Member of Council): I have very much pleasure in seconding the vote of thanks to the Society of Arts, which has been proposed by Mr. Macfarlane Gray in such suitable terms.

The motion, having been put to the meeting, was carried unanimously.

Admiral Sir Nathaniel Bowden-Smith, K.C.B. (Associate): It is my pleasure, gentlemen, to propose a vote of thanks to the Council for their services during the past year. Their good work is so apparent to you all that I need not occupy your time, but merely ask you to pass a hearty vote of thanks to the Council for their services during the past twelve months.

Mr. G. W. Manuel (Member): Gentlemen, it is with very great pleasure I have to second the vote of thanks to the members of the Council of this Institution. They give their labours unstintedly. Many of them connected with business come here voluntarily and do the work, and I think the members should give the Council of this Institution a hearty vote of thanks. They have placed before us all these valuable papers; it is the Council who select these papers, and there is a great deal of thought required in providing a programme of such an interesting description. It is with very great pleasure that I ask you to pass this vote of thanks.

The motion, having been put to the meeting, was carried unanimously.

Mr. Henry Morgan (Vice-President and Treasurer): Sir John Hay and Gentlemen, I have very great pleasure in expressing to you the thanks of the Council for the kind way in which you have expressed your appreciation of their labours in the past year. You are aware that those labours have been in one respect of no common character. They have had the very onerous duty of selecting a new President. The success with which they have carried out that object has been expressed to you in Sir John Hay's speech, and it is perfectly manifest to us all who have attended this meeting. We have before us in the coming year another, and, if possible, more onerous duty. I refer to the election of a

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new Secretary. When we meet here this time next year, we shall see somebody else occupying the chair that our esteemed friend has occupied so efficiently for the last twenty-two years. It is not my business or desire to speak in laudation of the loyal service which he has rendered, because it has been done better on another occasion; but I would call attention to the fact that the Council have the very difficult task imposed upon them of filling the position which he is about to vacate, and have in fact already entered upon the necessary steps, and I trust we shall carry out that duty with as much satisfaction to you as you have evinced with regard to the election of our President. We know that we shall have your assistance in doing so, and every help that lies in your power to give. We have to look forward to the meetings in Glasgow in June, and a further function later on in the autumn of the year in conducting our section of the Glasgow Exhibition. In those matters I trust we shall meet with success, and I assure you that the Council will be ready, as they have been in the past, to use their utmost endeavours to bring those undertakings to a successful issue. I trust, when we meet next year, we shall deserve the same cordial vote as you have now accorded us.

The Chairman (Admiral the Right Hon. Sir John Dalrymple-Hay, K.C.B., D.C.L., F.R.S., Vice-President): Your thanks to the President have already been voted, but this is the last occasion on which we shall see Mr. Holmes in this chair. Mr. Holmes has been our Secretary for twenty-two years, and as Secretary he has fostered this Institution as much as it was possible for any person in his position to do. He is now going to a higher appointment. He is going to Ireland as Chairman of the Board of Public Works there; but, in spite of the fact that the Government are anxious that he should go as soon as possible, he has, with that loyalty to this Institution which he has always shown, foregone a fortnight of his salary there and of his official duties, in order to remain here to complete the business that has taken place during this session. I therefore think that he deserves the most grateful thanks of the Institution, which I trust you will give him. I move a most hearty vote of thanks to Mr. Holmes.

Mr. George Holmes (Secretary), Sir John Hay and Gentlemen: I am deeply sorry that this is the last occasion on which I shall address you from this chair. I thank you from my heart for the kindness and the support which you have always shown to me during the twenty-two years I have been in your service. I am sure I can promise you that, if you will show that kindness and support (as I am certain you will do) to my successor here, whoever he may be, he also will endeavour to serve you as you would like to be served. I feel confident that this Institution will prosper always. It is doing such important work for the country and the world that it is certain to be well received wherever it goes. I foresee the greatest future for it, and wherever I go, and whenever I have a chance of serving you, even in the smallest degree, though I may be far away, you may be sure that I will do my best. I thank you again for all you have done for me.

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# PROCEEDINGS IN GLASGOW.

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SUMMER MEETINGS OF THE FORTY-SECOND SESSION

OF THE

# INSTITUTION OF NAVAL ARCHITECTS.

JUNE 25, 26, 27, AND 28, 1901.

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# INTRODUCTORY PROCEEDINGS.

THE Summer Meetings of the Forty-Second Session of the Institution of Naval Architects were held at the Corporation Galleries, Sauchiehall-street, Glasgow, on June 25, 26, and 27, 1901.

The President and Council of the Institution, after having been welcomed by the Lord Provost of Glasgow and the members of the Reception Committee in the Municipal Buildings, George Square, proceeded to the Corporation Galleries, where the Lord Provost took the chair.

The CHAIRMAN (the Hon. the Lord Provost, Samuel Chisholm, Esq., LL.D.): Lord Glasgow, my Lords, Ladies, and Gentlemen, I have the high distinction of occupying the chair for a few moments this morning, in order that I may, in the name of the Municipality and the citizens of Glasgow, offer the very heartiest welcome to the Institution of Naval Architects, which has this season chosen Glasgow as the seat of its Congress. I desire to assure you that we regard your visit to us at the present time with very great interest and satisfaction. Nothing that is connected with the building of ships can possibly be a matter of indifference to the citizens of Glasgow. We have long been too closely identified with the building of ships in the City of Glasgow, and on the Clyde which flows through it, to be indifferent to anything which is connected with that subject, and we therefore look forward with very great interest to your gathering in our midst at the present time, and to the discussion of the important subjects connected with naval architecture to which your attention will be directed. I have now very great pleasure in vacating the chair, and introducing to you one who needs no introduction to this Institution, having already been elected your President, my Lord Glasgow, who will deliver to you his opening address.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.) then took the chair.

The PRESIDENT: My Lord Provost, my first duty is a very agreeable one indeed; it is to express to you on behalf of the Institution of Naval Architects our sincere thanks for the kind welcome which you have just given us in the name of the Corporation in such eloquent terms. My Lord, my own experience of your hospitality during the late celebration of the Ninth Jubilee of the University of Glasgow is so recent that I am able to anticipate with interest the pleasure which the members of the Institution will receive during their visit to this city, from the Scottish welcome which will meet them, not only from the Corporation of Glasgow, but wherever they may go.

They will also, I trust, enjoy their visit to the interesting International Exhibition situated so picturesquely on the banks of the Kelvin, under the shadow of the stately buildings of the University,



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and they will, I have no doubt, experience the liveliest satisfaction in the inspection of the beautiful collection of paintings in the magnificent Gallery of Arts and Science which has been lately built—a gallery which has no superior, even if it has an equal, anywhere in the world.

My Lords and Gentlemen, you will all, I feel sure, be pleased to see among us so many of our friends from other countries—especially a large number of our German fellow members. We fully recognise the great impetus which has been given of late to shipbuilding in Germany, and we wish all success to the Naval Architects of that great country in their work.

This is not the first occasion on which our Institution has found itself on the banks of the Clyde. It was in 1889, I think, that we paid our last visit here. On that occasion our much-esteemed President, the Earl of Ravensworth, in the course of his address, spoke of Glasgow as at once the birthplace, the cradle, the nursery, and the high school of steam navigation. This is great, but not undeserved, praise; and, with your leave, I purpose to show you, in the remarks I am about to make, the admirable manner in which the Clyde Navigation Trust carries out the duty that is incumbent upon it, of keeping the harbour of Glasgow abreast of the times, and of accommodating the vast and ever-increasing volume of trade that comes to the Clyde. I will also show you the work our shipbuilding yards are doing, and, generally, how prosperous is this great centre of industry.

The year 1900 has been the record year of shipbuilding on the Clyde. Of the 541,031 tons launched during that year in Scotland, no less than 444,609 tons were built on the Clyde, leaving only 49,429 tons for the whole of the rest of Scotland. But that is not all. In England last year the total tonnage launched was 961,426 tons, a very little more than double that built on the Clyde, so that the Clyde shipbuilding for the year is almost half that of the whole of England.

This is a position in shipbuilding of which Glasgow has every reason to be proud; let us hope that the trade on the Clyde will continue to prosper, and that nothing may arise to imperil its well-being or to check its development.

Turning to the progress made by the Clyde Navigation Trust in enlarging the harbour of Glasgow since your visit twelve years ago, I have much pleasure in laying before you the following figures:—

| | In 1889. | In 1901. | Increase. |
|---|---|---|---|
| Length of quayage ... | 10,839 yards, or 6 miles 275 yards | 15,115 yards, or 8½ miles | 2¼ miles |
| Water area... ... ... ... | — | — | 52 acres |
| Revenue of the Trust ... ... | £287,953 | £441,419 | £153,466 |
| Customs revenue ... ... ... | £1,012,051 | £1,952,910 | £940,859 |
| Total tonnage exports and imports | 3,723,058 tons | 7,215,368 tons | 3,492,310 tons |
| Total net registered tonnage ... | 3,118,515 tons | 4,361,597 tons | 1,243,082 tons |

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But the work of the Clyde Navigation Trust has not been confined to increasing the length of their quays and the acreage of their harbour. In 1889 they possessed two exceedingly fine graving docks; but, not considering that sufficient, they proceeded to make another dock larger than the others. In graving dock No. 3, the Trust now possess a dock believed to be second to none in existence. Its length is 880 ft., or 130 ft. longer than the well-known Prince of Wales' Dock at Southampton; its width at the top of the entrance is 82 ft., and the same at the bottom, and the depth on the sill at ordinary high water is 26 ft. 6 in. It is thus large enough and to spare for the largest vessel afloat, or likely to be afloat.

The Trust has not neglected the deepening of the river, which is continually being proceeded with. In 1889 the Clyde was navigable by vessels drawing 24 ft.; this year vessels drawing 26 ft. can go up or down in one tide without difficulty.

From what I have said, it is, I think, patent to all that everything is being done to keep the harbour of Glasgow in the condition of efficiency in which all interested in its welfare would wish to see it.

There are always a large number of men-of-war building in the numerous shipbuilding yards on the Clyde. At present there are nine, of which eight are first-class armoured cruisers, and one is a torpedo-boat destroyer. Of the former, two are of 14,100 tons and 30,000 I.H.P., one is of 12,000 tons and 21,000 I.H.P., and five are of 9,800 tons and 22,000 I.H.P. They are as follows:—

| | Tons. | I.H.P. | |
|---|---|---|---|
| *Leviathan* | 14,100 | 30,000 | John Brown & Co. |
| *Good Hope* | 14,100 | 30,000 | Fairfield Company. |
| *Bacchante* | 12,000 | 21,000 | John Brown & Co. |
| *Monmouth* | 9,800 | 22,000 | London and Glasgow Company. |
| *Cumberland* | 9,800 | 22,000 | " " " " |
| *Bedford* | 9,800 | 22,000 | Fairfield Company. |
| *Donegal* | 9,800 | 22,000 | " " |
| *Berwick* | 9,800 | 22,000 | Beardmore & Co. |

The *Cressy* and *Aboukir*, sister ships to the *Bacchante*, have lately left the Fairfield Company's works; and the *Sutlej*, a similar ship, has also recently left Messrs. John Brown & Co.'s works. All these cruisers, except the *Bacchante* and her sister ships, are to be 23-knot boats.

Of torpedo-boat destroyers building, there is at present only the *Arab*, 430 tons, 8,600 I.H.P., by John Brown & Co.; but the *Falcon*, *Ostrich*, *Thorn*, *Tiger*, and *Vigilant* have recently left the Clyde.

It may interest you to hear of a new departure which is being made on the Clyde. The screw steamer *King Edward* is the first of a new class built for the Clyde passenger traffic. She is driven by Parsons' patent turbine machinery, having five propellers on three shafts, one on the centre shaft and two on each of the side or wing shafts, revolving at a speed of 1,000 revolutions a minute. The particular feature of the engine is the almost entire absence of vibration and the economy of space in the engine department. She does not differ much externally from other Clyde passenger boats of later types in appearance, except that she has no paddles nor projecting sponsons. She possesses excellent accommodation, and her certificate allows her to carry 2,000 passengers. She has had only

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one trial, and that was not to test her speed; it is, therefore, premature to say more about her than that, if she turns out the success that the syndicate which has built her expects, she will create a revolution in the coast passenger-carrying trade, which on the Clyde is now carried on (inside of Loch Ryan) by 37 steamers, 27 of which belong to the three local railway companies.

Our last Summer Meeting was at Newcastle-on-Tyne, and the late Lord Armstrong, then in his ninetieth year, took a deep interest in the preparations which were made for the reception of the Institution, whose members he entertained at his seat at Cragside. I beg that you will bear with me while I allude to the great loss which the Institution has so lately sustained. By the death of Lord Armstrong, we are deprived of one of the keenest of intellects, and of a man who, in his day, has been most distinguished in his profession, in which he always displayed marvellous practical sagacity. His great creation was the Elswick Works, where his own energy carried out the most far-reaching changes in our armaments, and the shipbuilding works of his firm have produced some of the most perfect specimens of men-of-war for both our own and other navies. He resigned the post of Under-Secretary for War many years ago, because he felt that he could be more useful to the country in an independent position. Though his health was uncertain in his later years, yet at the age of eighty-eight he completed and published an abstruse work on science, which demonstrated that his powers of mind were not abated. He has passed away at a very advanced age, and his native town of Newcastle will long cherish his memory and value the great gifts with which his genius has endowed it.

I feel that I cannot conclude my address without some allusion to my old and esteemed friend, the late Lord Inverclyde, one of the most distinguished of British shipowners. Lord Inverclyde was a representative type of Scotsman, and a man of great business aptitude. His life was associated with the history of that great steamship company, the Cunard, of which he was so long the head and the moving spirit. He was ever foremost in maintaining the efficiency and the supremacy of the British Mercantile Marine.

He performed eminent service to the country in being the first to draw the attention of the Government to the great advantage to be derived from fitting and adapting merchant ships so as to be capable of being used for warlike purposes, and also in recommending the arrangements by which the Government authorities can command the services of a fleet of transports when required. For his advocacy of these measures he earned the esteem and the respect of all the leading Naval officers of the day, and he is mourned by all who appreciated his high integrity and his public spirit.

My Lords and Gentlemen, I feel that I have trespassed too long on your forbearance, and I must now bring my remarks to a close.

It is with feelings of pride and satisfaction that I have been able to lay before you such a record of the progress which has been made by this great and prosperous port and district since our visit to it twelve years ago. Marvellous indeed has been its whole history when we remember that one hundred years ago Glasgow was a town of only 12,500 inhabitants; indeed, unless we turn to the new cities of the United States, it would be difficult to point to any city which can boast of so large and rapid a growth.

We in the West of Scotland are proud of our commercial capital. I am sure that all present wish it continued prosperity, and that you will all cheerfully re-echo the aspiration which has been

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adopted as the city motto, and is engraved under the City Arms, with which aspiration I conclude my address—"Let Glasgow Flourish!"

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, before we commence the business of this meeting, I have an agreeable duty to perform. You are aware that we have had the great misfortune to lose our most excellent and esteemed Secretary, Mr. Holmes, whom we are glad to see present to-day. While we regret that he has left us, we wish him every success and happiness in his new sphere. But, gentlemen, it is a case of "The king is dead, long live the king!" We have present with us his successor, Mr. Dana, who has been chosen from among 150 odd different candidates for the appointment. The Council consider that in Mr. Dana they have made the very best choice that they possibly could, and I think that we have got a Secretary with whom it will be a pleasure for us to work, and whom it will be a pleasure for the Members of the Institution to meet. I will now introduce Mr. Dana to you, and I beg to congratulate him on his appointment, and you on having got him as your Secretary.

The SECRETARY (Mr. R. W. Dana): My Lords, Ladies, and Gentlemen, allow me to thank you most heartily for the kind manner of my reception, and for the gracious words that have fallen from Lord Glasgow. I am sure I shall do everything in my power to further the interests of the Institution, and to justify the selection that has been made.

The Right Hon. Lord BRASSEY, K.C.B., D.C.L. (Past President): My Lord Glasgow, Ladies, and Gentlemen, at the advanced age which I have reached, the inducements are very powerful to coil down and do nothing more. That seems an unworthy course, if one may venture to hope that the country will gather some fruit from one's labours. I am here, therefore, to submit for the consideration of this powerful body, proposals which, if they receive support here, are likely to carry much greater weight than if they were presented on my sole authority, for the consideration of those who are responsible.

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# MERCANTILE AUXILIARIES.

By the Right Hon. Lord Brassey, K.C.B., D.C.L., Past President.

[Read at the Summer Meetings of the Forty-Second Session of the Institution of Naval Architects, June 25, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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In bringing forward proposals for a more vigorous policy in relation to mercantile auxiliaries, I return to a subject in which, following the lead of Sir Nathaniel Barnaby and other authorities, I have long been deeply interested.

The history of the question may be briefly traced. The earliest contracts for the conveyance of the mails required that postal vessels should be suitable for conversion into armed cruisers. In 1853, upon the recommendation of the Committee on Postal Contracts, those wise stipulations were withdrawn. It is a notable circumstance that, at the same date, iron was condemned as a material for the construction of fighting ships. After the lapse of a quarter of a century the subject was taken up by Sir Nathaniel Barnaby. His views were presented for the consideration of this Institution in a paper read in the Session of 1877 on the fighting power of the merchant ship. Sir Nathaniel Barnaby insisted on the impossibility of providing adequately for the protection of our vast commerce with regularly built vessels of war. It was impossible to say what number of fast cruising ships would be sufficient for England, seeing that we should never know in what part of the world we should be attacked. It was necessary, therefore, to supplement—and largely to supplement—our regular cruisers with auxiliary vessels. Sir Nathaniel Barnaby's recommendations found powerful supporters.

Admiral Sir Frederick Gray spoke as follows: "Having been at the Admiralty and having felt the difficulty of providing, even in peace time, the force necessary to fulfil the various duties devolving on our ships of war, I think it would be utterly impossible to provide sufficient protection for our Mercantile Marine in time of war. I believe that the merchant steamers of England, if strengthened and fitted as proposed by Sir Nathaniel Barnaby, would be very useful. I do not say that they would take the place of men of war, but they would be most useful as auxiliaries." In the same discussion Sir Spencer Robinson, then Comptroller of the Navy, expressed himself in these

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words: "Those splendid ships which pass between the United States and England with very great speed and a coal-carrying capacity far exceeding the coal-carrying capacity you can give to any unarmoured man of war, could be made capable of carrying such an armament as would protect them from the attacks of vessels of a similar class to their own."

These views of the naval advisers of the Admiralty, vainly urged a generation ago, have been fully shared, and practically adopted, by the naval advisers of foreign Powers. I may quote the remarks of Admiral Fournier, in his able pamphlet, entitled, "La Flotte nécessaire." "As types of fast cruisers for the destruction of commerce, I know of nothing which more fully meets the requirements than those magnificent Transatlantic steamships, the *Lucania* and *Campania*, capable of maintaining a speed of 22 knots an hour, with extraordinary uniformity. The *New York*, *Paris*, *St. Louis*, and *St. Paul*, and the English ships *Majestic* and *Teutonic* possess the same qualities, though in a somewhat less degree of perfection. Such ships will, in my view, be the destroyers of commerce in the future."

While their chief naval and technical advisers were endeavouring to obtain the assent of the Board of Admiralty to some practical action, the subject was not neglected out of doors. In 1878 a paper was published in the *Nautical Magazine* by the late Lord Inverclyde. I quote his weighty words, as true to-day as when they were written: "There never was a time in the history of this country when the subject of the efficiency of the Royal Navy occupied a position of greater importance than it does at present. Our risks lie in the fact that the fleets of other nations are fast becoming powerful and reliable; and whilst no navy can numerically approach that of this country, yet there are nations in Europe whose fleets combined would undoubtedly give us enough to cope with. How then can we stride ahead as the greatest maritime Power, and hold our own against the fleets of the world? Not by being satisfied with increasing the strength of the Navy proper, which, owing to the prodigious cost of modern war vessels, can only be done in a comparatively small degree. But what cannot be accomplished in that direction can be attained by other means ready to our hand, and that is by utilising the vessels of the Mercantile Marine."

At the date of Lord Inverclyde's paper the cost of our most powerful cruisers was under a quarter of a million. The cost of the first-class cruisers we are now building is more than threefold greater. The practical steps which Lord Inverclyde recommended to the Admiralty are briefly described in his paper. There was wanted a scheme by which the advantages of the vast fleet of merchant steamers now belonging to the country should be reserved for our special requirements, and it is due to our Naval authorities to admit that they recognised that there were numerous British vessels which could easily be converted into cruisers. There was, however, one fatal

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flaw in the Admiralty plan. They wanted to have the use of the ships without paying an adequate consideration! As Lord Inverclyde put it: "It was not to be supposed that a position for their vessels on the select list would be a sufficient inducement to comply with Admiralty requirements, and to incur the expense involved in executing the requisite alterations. War being only a contingency more or less remote, a retaining fee must be offered." It was proposed by Lord Inverclyde that subsidised steamers should be built to meet certain requirements, including increased bulkheads and water-tight compartments. The ships should be manned by seamen of the Royal Naval Reserve, who should be thoroughly trained in gunnery at the respective home ports of the companies or owners.

The subject of mercantile auxiliaries continued to attract the attention of our highest authorities on naval administration. In 1880 Sir Donald Currie read an exhaustive paper at the United Service Institution. He referred to the general increase in naval preparations. France had increased her navy; Germany and Russia were making large strides in the direction of more powerful naval forces. Quite recently the Russian volunteer fleet had been originated by Prince Dolgorouki, the Governor-General of Moscow. He addresssd himself to the wealthy merchants of that ancient capital, appealing to their patriotism, and a Volunteer Cruisers' Fund had been raised. Sir Donald Currie submitted a scheme for the retention of swift merchant cruisers by an annual subsidy.

I may refer to another shipowner, most eminent in that branch of enterprise with which he was connected. I refer to the late Mr. Ismay. In the evidence which he gave before the Royal Commission on coaling stations, he truly said: "When a company has not been doing well, and has got heavy bills running, which it cannot meet, the temptation to shut its eyes to what would be the ultimate destination of ships sold to foreigners in a crisis would be very great. At the commencement of the Russian scare, on the occasion of the Penjdeh incident, great temptations were offered to the owners of the White Star steamers running between San Francisco and Japan, and it was not too much to say that our whole commerce in the Pacific would have been transferred to the flag of the United States, if that offer had been accepted." The British ownership of a magnificent steamship is a slight national tie. It binds to no national service while it exists, and it may be broken without warning, at the will of the owners. These considerations may be pleaded as a strong argument in support of the policy of binding all our finest vessels to the service of the State as mercantile auxiliaries.

Passing on to the later authorities, the practicability of so constructing merchant steamers as to render them readily available for war purposes was discussed by



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Professor Biles in a paper read at an engineering conference, held under the auspices of the Institution of Civil Engineers, in June, 1899. Arguing from the results of arming and fighting the mercantile cruisers of the United States Navy, Professor Biles takes the view that such vessels are not unable to cope with thorough-bred warships. The experiences of modern sea fights point to the conclusion that a ship is more likely to be disabled by her crew being driven from their guns than to be sunk by the effects of shell fire. The issue of a fight between a warship and a merchant ship may not turn on the relative efficiency of the internal subdivision, but on the protection of the guns. By placing the guns in a box battery, as the Americans did, a considerable number of guns could be as well protected in merchant ships as in first-class cruisers. Professor Biles holds it to be practicable, with due consideration in the early stages of design and construction, to so protect machinery and armament, that the merchant ship need not be much, if at all, inferior to many warships.

The policy of liberal subsidies to the Mercantile Marine has given to the merchant navies of foreign Powers a decided advantage in the possession of the types of vessels most suitable for naval purposes. The Germans have taken the lead. They have two ships now running, the *Deutschland* and the *Kaiser Wilhelm*, of 14,000 tons and 15,000 tons respectively, which exceed in speed by nearly 2 knots our best ships. No vessel now building for the British flag will rival in speed the *Kaiser Wilhelm II.* and *Kronprinz Wilhelm* under construction in Germany.

Of ships capable of a regular sea speed of over 18 knots, France has 4, Germany 8, and Great Britain 10.

Taking vessels above 3,000 tons only, Lloyd's statistics, as quoted by Professor Biles, give the relative position as follows :—

| | British. | Foreign. |
|---|---|---|
| 20 knots and over ... ... | 6 | 6 |
| 19 to 20 knots ... ... ... | 1 | 11 |
| 18 to 19 knots ... ... ... | 9 | 4 |
| 17 to 18 knots ... ... ... | 22 | 18 |
| 16 to 17 knots ... ... ... | 17 | 18 |
| | 55 | 57 |

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The latest returns give the relative position as follows:—

OCEAN-GOING STEAMSHIPS OVER 3,000 TONS.

OVER 20 KNOTS.

| | |
|---|---|
| British ... ... ... ... ... ... | 7 ships. |
| French ... ... ... ... ... ... | 2 „ |
| German ... ... ... ... ... ... | 5 „ |
| Russian ... ... ... ... ... ... | 1 „ |

Since the year 1893, when we built the *Lucania* and *Campania*, the *Oceanic* is the only vessel added to the British list. Since 1897 Germany has built four ships, France two, and Russia one ship. Seven ships have been built under the foreign flag, as against one under the British flag.

Comparing the 3,000 ton ships of lower speeds, the relative position is as follows:—

19 TO 19½ KNOTS.

| | |
|---|---|
| France ... ... ... ... ... ... | 2 ships. |
| Germany ... ... ... ... ... ... | 2 „ |
| Russia ... ... ... ... ... ... | 4 „ |
| Spain... ... ... ... ... ... ... | 1 ship. |

18 TO 18½ KNOTS.

| | |
|---|---|
| British ... ... ... ... ... ... | 1 ship. |
| Germany ... ... ... ... ... ... | 3 ships. |

In ships steaming under 18 knots, those flying the British flag show a decided superiority.

It is an ominous fact that, while we equal the combined merchant navies of the world in aggregate tonnage, of the 157 ships of 16 knots and over, less than one-half the number are under the British flag. If we examine the lists of mercantile auxiliaries of the several Powers, we find that France has 32 vessels, the latest additions, the *Lorraine* and *Savoie*, having a displacement of over 11,000 tons. The speeds of the French subsidised steamers have been constantly increasing from the 15 knots of twenty years ago, to the 17, 18, and 19 knots of modern types. The armaments provided for each ship include seven 5·5 in. guns and smaller quick-firers. In numbers, the German auxiliaries do not compare with the French. There are, however, six ships of the first class, which, as it has already been observed, hold the record in the international competition on the North Atlantic. The armaments prepared for the German mercantile auxiliaries include eight 5 in. guns, four 4·7 in., four smaller quick-firers, and fourteen machine guns. The Russian fleet of auxiliary steamers consists of 25 vessels. In displacement, eight of these ships exceed 10,000 tons with speeds of 19½ to 20 knots. The British list of reserve merchant cruisers compares unfavourably with those of foreign Powers. It consists of 29 vessels, none being fitted with special protective arrangements.

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With these comparisons before us, we must regret that so little heed should have been paid to the counsels of the able men to whom I referred in the opening of this paper. How little, is attested by the fact that the amount payable to the owners of our reserve merchant cruisers under the Naval Estimates now before Parliament is £63,000, out of a total of £32,000,000 which Parliament is asked to vote for the Navy.

While we have taken no adequate measures to create a fleet of mercantile auxiliaries by the appropriation of moneys directly voted from Navy Estimates, we have been sparing in our grants for postal services. The amounts paid have been computed by Mr. Henniker-Heaton, M.P., as follows :—

| Countries. | Money. | Total Value of the Foreign Trade. |
|---|---|---|
| France ... ... ... | £1,143,000 | £300,000,000 |
| Germany ... ... | £1,000,000 | £313,000,000 |
| Russia ... ... ... | £251,000 | £111,000,000 |
| Italy ... ... ... | £400,000 | £182,000,000 |
| Great Britain ... ... | £637,000 | £750,000,000 |

The latest statement issued by the Foreign Office gives the amounts paid for bounties and subventions as under :—

FRANCE, IN 1900.

| | |
|---|---|
| For construction of new ships ... ... ... ... | £196,000 |
| For navigation, tonnage, and mileage ... ... ... | 452,000 |
| | 648,000 |
| Postal subsidies ... ... ... ... ... ... | 1,067,000 |
| | £1,715,000 |

RUSSIA ... ... ... £318,000

(This includes £127,500 to the Volunteer Fleet.)

The total payment for British postal services is less than £700,000, although the value of our foreign trade exceeds the aggregate amount of that of France, Germany, and Russia combined.

And now, let us ask ourselves what steps it is practicable to take in furtherance of the policy which it has been the object of the present paper to recommend; and, first, what is the proper sphere of action of the mercantile auxiliary? It was described by

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Lord George Hamilton, in moving the Navy Estimates for 1889. They would be employed in dogging the footsteps of a foreign rover, and embarrassing a foe; they would act as the scouts of the fleet. The difficulty of keeping touch with a squadron has been strongly impressed in recent manœuvres. As scouts and patrol vessels their long coal endurance would give to the mercantile auxiliaries an advantage over regularly built vessels of war of even tonnage. Such being the rôle of the mercantile auxiliary, the qualities required are speed, coal endurance, internal subdivision up to Admiralty requirements, strength of hull. Guns must be carried at such a height above water as to be of service in a seaway. Lastly, there is the absolute requirement of protection from the shot and highly explosive shell of quick-firing guns. The power of modern armaments to reduce unarmoured superstructures to wreckage, to disable guns, and annihilate guns' crews has been signally shown in the battles of Manila and Santiago.

If the requirements were carefully considered in the original design, it would be possible to give as much, or nearly as much, protection to the mercantile auxiliary as to the regularly built cruiser. Mr. Peskett, of the Cunard Company, has recently offered some practical suggestions on this subject in a lecture delivered at Liverpool. Mr. Peskett is a strong advocate for subsidies to mercantile auxiliaries, conditional upon the adaptation of the ships to the requirements of naval warfare. I may quote from a précis of his lecture :—"Merchant ships of the *Campania* or *Saxonia* class could actually be built lighter than they are under the present system, if they were built with one very strong deck, such as a protective deck with sloping sides, or with a deck of cellular form. The disposition of material in some of our large steamers is not perhaps in strict accordance with the best designs our naval architects could produce, but entirely due to the requirements of owners, and the various registration societies. I should say that a cruiser's hull, with protective deck, is lighter in proportion to her displacement than that of many of our first-class passenger steamers.

"Taking into consideration the fact that our supremacy depends on the efficiency of our Naval and Mercantile Marine, a committee of Admiralty officials, shipowners, and shipbuilders should be formed to discuss the best method of constructing a combined Naval and Mercantile Marine, and to consider whether ships could be built as merchant cruisers, with protective decks, ram stems, machinery, and steering gear below the water-line, and still be able to carry enough passengers and mails, which with a reasonable subsidy would make the ships remunerative to owners.

"These ships would have to be permanently mounted with light guns, racer plates for heavier armaments being built in the ships during construction, the heavier guns and mountings being kept at ports of call, and made to suit the various ships of any particular fleet."

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It has always been recognised that mercantile auxiliaries cannot be effective unless specially designed for conversion into cruisers. In the discussion at the Institution of Naval Architects on Mr. Barnaby's paper already referred to, Sir Edward Reed proposed that shipowners about to build large and swift merchant vessels should be invited to submit their plans to the Admiralty. The cost of any alterations which they might be willing to make in order to adapt these vessels for the emergency of war service should be met by a grant from the Government.

It is the main object of the present paper to urge the adoption of valuable and practical recommendations, which have been too long neglected. It is the fixed resolve of the people, and, perhaps, the first duty of British statesmen, to keep the Empire secure from attack, and to give protection to the commerce on which our existence depends. In pursuance of this policy we have more than doubled the expenditure under Navy Estimates, and still we seem to fall short of the full requirements for the naval defence of the Empire. It would be impossible to fix a limit to the number of cruisers required for the protection of a commerce which extends over every sea. The construction of cruisers has absorbed in recent years a large proportion of the shipbuilding votes. But when the cost of the first-class types falls little short of that of the battleship, the numbers we can build are all too few for the work they might be required to do.

We cannot cut down the expenditure on battleships. If, therefore, our regular built cruisers are fewer than we could wish, we must look to our own Mercantile Marine, and, out of the abundant materials we there find ready to our hands, we may organise a supplemental fleet of armed cruisers such as no other State can furnish. The Admiralty should utilise these resources by liberal subsidies. The standard of requirements should be high. The speed should not be less than that of the *Deutschland*—let us say, 22 knots at sea. Mercantile auxiliaries should be protected by a deck or belt of Harveyised armour. The necessary armaments should be in readiness. Calculations of cost can hardly be given in a paper in which nothing more is attempted beyond suggesting a policy. This, at least, is certain, that the cost of the adaptations and protective arrangements necessary in a mercantile auxiliary will be small in comparison with the first cost of a regularly built vessel of war. For a first-class cruiser we may take the cost at three-quarters of a million. Allowing 3½ per cent. on the money invested, adding 6 per cent. for depreciation, and 1 per cent. for insurance, we have in round figures for the first cost an annual writing down charge of £70,000 a year. In addition, there is the cost of maintenance, which, whether in commission or in reserve, will certainly be considerable.

Allowing for the protective arrangements of the mercantile auxiliary the liberal sum of £50,000, and a writing down charge on this amount of 10 per cent., and taking

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10s. per ton displacement for the annual retainer, we have, for a mercantile cruiser of 10,000 tons, an annual charge of £10,000. We must further take into view the economy resulting from the maintenance of the mercantile auxiliary by the shipowners, as against the maintenance of the man-of-war in our Royal yards.

In conclusion, I claim that it has been clearly shown that we can have many auxiliaries for the cost of one cruiser; and these auxiliaries may have effective protection. If not equal to the ship of war as combatants, they will be superior in coal endurance and probably in speed for long distances. They would be the scouts of our fighting squadrons. They would protect our commerce from interruption by the auxiliary vessels of a hostile Power.

While the building of cruisers for the Navy should be continued, the resources we possess in the marine which our maritime enterprise has created in extent practically without limit, should not be neglected. Here, in Glasgow, I look for powerful support to a policy not now presented for the first time to the consideration of the Institution of Naval Architects. It has been advocated for many years by your ablest men, it is being steadily pursued, to their signal advantage, by every naval administration: that we are seriously lagging behind has been shown by the figures which I have quoted. If the expenditure on auxiliary cruisers were raised from £60,000 to £600,000 a year, in a few years we should be enabled to provide no ineffective protection for our vast trade.

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# CONVERSIONS AND RE-ARMAMENT OF SHIPS ON THE EFFECTIVE LIST.

By the Right Hon. Lord Brassey, K.C.B., D.C.L., Past President.

[Read at the Summer Meetings of the Forty-Second Session of the Institution of Naval Architects, June 25, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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Giving expression to the national resolve that the Navy shall be strong enough to ensure the integrity of the Empire, Parliament has in recent years voted appropriations for the construction of new ships with no unstinting hand. We have not made full use of the means we possess of reinforcing the Navy by conversion and repair. In a period of extraordinary developments in the manufacture of guns and armour, many sound and seaworthy ships require improvements to keep them up to date.

The policy I desire to recommend for the British Navy has been actively pursued by foreign Powers. I might refer to the conversion of ships belonging to the Austrian Navy; to the rebuilding of monitors in the United States; and, more particularly, to the extensive work of reconstruction which is in progress under the French Naval Administration. In France, as the result of an exhaustive examination of the subject, a scheme was approved in 1896 for the reconstruction of five coast defence vessels. In 1897 a similar scheme of reconstruction was approved for no less than ten battleships.

The alterations which have been carried out include the cutting down of the heavy superstructures with which the French ships of the most recent types have been encumbered; the fitting of Belleville and Niclausse boilers; the modernisation of machinery; and the arming of the ships with guns reduced in calibre and lighter in weight, but having a more commanding range than the earlier patterns.

The ships with which it was decided to deal include the *Formidable*, *Baudin*, *Courbet*, *Redoubtable*, *Devastation*, *Hoche*, *Marceau*, *Amiral Duperré*, *Neptune*, and *Magenta*. The oldest ship in this list is the *Courbet*, launched in 1881; the latest is the *Magenta*, launched in 1890. The ships already converted are the *Formidable*, *Courbet*,

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*Redoubtable*, *Devastation*, and *Marceau*. The list of coast defenders includes the *Terrible*, launched in 1881; the *Caïman* and *Indomptable*, both launched in 1885. These three ships are actually in hand. To complete the list, we have the *Requin*, launched in 1885, and partially rebuilt in 1896; and the *Furieux*, launched in 1883.

The refit of the *Amiral Duperré* includes a new boiler, and the substitution of four 6·4 in. quick-firing guns in a central casemate for the 13·3 in. gun formerly carried in the barbette amidships. The refit of the *Neptune* comprises the substitution of Bellevilles for cylindrical boilers, and the necessary adaptations of engines for working at higher pressures. The ponderous superstructure will be removed.

In the coast defence ships the work of reconstruction has been carried further than in the battleships. In the *Requin*, of 7,800 tons, and built fifteen years ago, the armour on the belt has been reduced from 19·5 to 9 in. For the 16·5 in. guns formerly carried, two 10·8 in. guns have been substituted. Lightened to the extent of over 600 tons, the ship will be more seaworthy and better protected. The *Furieux* is to be fitted with Belleville boilers, and two 9·4 in. guns in closed turrets will be substituted for the two 13·3 in. guns formerly mounted in the barbettes; the ship will be lightened by 246 tons.

The total expenditure originally authorised for the refits and reconstruction recently undertaken in the French Navy was, in round figures, a million sterling. It goes without saying that this figure may be considerably exceeded. The expenditure will be amply justified. No less than fifteen ships, which were becoming obsolete in armaments, armour, boilers, and machinery, will have been brought up to date. A policy pursued with such successful results in France should be good for our own Navy.

Turning to Germany, the reconstruction of a considerable group of armoured ships has given a powerful reinforcement to the navy. The coast defence armoured ship *Hagen*, which was one of the German vessels sent to Spithead for the late Queen's funeral, had just undergone a complete reconstruction. She belongs to the *Siegfried* class, built for local defence, and with a restricted coal endurance. The *Hagen* was cut in two amidships, and lengthened 27 ft. The coal capacity was thus increased from 225 to 580 tons. Four locomotive boilers have been replaced by eight Thornycroft water-tube boilers, giving 5,250 H.P. and a speed of 15 knots. The armament has been augmented, and now consists of three 9·4 in., ten 3·4 in., and six 1·4 in. guns, as well as machine guns. The torpedo armament now consists of 18 in. torpedoes, and the above water tube has been armoured. The armour on the conning tower has been increased to nickel steel 7 in. thick. It is intended that all the ships of the same type, seven in number, shall undergo a similar conversion.



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A member of Lord Northbrook's administration naturally looks to the ships of that period. Would that we could pronounce them as perfect by the light of recent developments as we thought them at the date when they were projected. Our six ships of the *Admiral* class were highly commended by competent critics. They steam well, and carry a fair coal supply. They are powerfully armed, and afford steady gun platforms. In other features they are less satisfactory. Since the *Admiral* class was designed, now nearly twenty years ago, we have seen a striking advance in the power of attack by quick-firing guns, and in the power of resistance by armour. Modern armour is equal to twice its thickness in the compound armour of the *Admiral* class, while modern guns have greater range and rapidity of fire and flatter trajectories. The total want of protection for the secondary armament of the *Admiral* class, always a weak point, has become a graver fault, in view of the increased power of attack on an unarmoured battery by quick-firing guns firing high explosives.

The *Admiral* class would gain greatly in fighting efficiency, if their batteries were protected. To give protection to the batteries, weight must be saved in other parts of the ship. In the barbettes two pairs of 28-ton guns might be substituted for the 12 in. guns at present mounted. The compound armour on the barbettes should be replaced by a more modern type of less thickness. Ships of the *Admiral* type, with the improvements which might be introduced, would be a match for many vessels which will long be retained on the lists of foreign Powers.

Pursuing a similar policy, the belted cruisers of the same period should be taken in hand. The programme recommended to Parliament by Lord Northbrook included seven ships of the *Aurora* type, of 5,630 tons. They are powerfully armed. They have a measured-mile speed of eighteen knots, and a coal endurance of 8,000 miles at ten knots. Within the limits of displacement available, it is simple justice to Sir Nathaniel Barnaby to say that the highest combination of qualities was obtained. By a liberal expenditure, judiciously applied, the *Aurora* class might be brought up to such a standard of fighting efficiency as would keep them on the list of "effectives" for many years. The improvements urgently required are increase of length, more freeboard forward, and greater protection by armour on the sides amidships. The necessity for increased length is clearly seen, when we look to the French types of approximately similar dimensions. Our belted cruisers, with a displacement of 5,600 tons, have a length of 300 feet. The French cruisers, both armoured and protected, have a conspicuous advantage in length. As examples, we may quote *Amiral Charner*, and two sister ships, displacement 4,792 tons, length 348 ft.; *Pothuau*, displacement 5,360 tons, length 370 ft.; *Dupuy de Lôme*, displacement 6,406 tons, length 374 ft.; *Catinat*, and two sister ships, displacement 4,065 tons, length 330 ft.; *Jean Bart*, and two sister vessels, displacement 4,100 tons, whose lengths vary in the

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different designs from 326 ft. to 346 ft.; *Cécille*, displacement 5,933 tons, length 378 ft. The lengthening of the belted cruisers would render many improvements possible. A large area of side might be protected, as, in the case of the French *Dupuy de Lôme*, with modern armour. Higher speed would be obtained with the same horse-power. The coal supply would be more ample. The bow might be raised—an important advantage in steaming against a head sea.

Having indicated a general idea, a non-professional administrator must leave the practical application to constructors and shipbuilders. In the case of ships built by contract, the work of conversion might be done by the firms that constructed the ships. Sir James Williamson, the Director of Dockyards, recently said, "The day is coming when the Admiralty must look to private firms to do repair work for the Navy, and I do not think they could do better than bring the ships back to the firms who built them." Allowing a wide margin, an expenditure not exceeding that of a single battleship should effect immense improvements in the six ships of the *Admiral* type. An expenditure not exceeding the cost of a single first-class cruiser should give us seven converted armoured cruisers which for many years would be capable of rendering effective service.

Expenditure on re-armaments, and on the fitting of new machinery for the older types of battleships, is not viewed with favour by constructors and shipbuilders. It must always be a more grateful task to build from original designs, the creation of their own science and invention, rather than to modernise ships which are becoming obsolete. Within prudent limits, no expenditure would yield more immediate results. The time taken for conversion need not exceed one-fourth that required to build a new ship. In war with a first-class Power, the sphere of action of our converted ships would be found in distant waters, and it is important that our flag should be carried worthily. The newest ships would be found in squadrons near home, in the Mediterranean and the Channel. The French are wisely making liberal appropriations for reconstructions, conversions, and refits. We throw away resources we possess ready to our hand, if we delay to follow on the same lines.

In advising as to the reconstruction and repair of ships, a consultative committee, on which the executive and professional officers of the Navy would be represented, might do good service.

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## DISCUSSION.

Admiral the Hon. Sir E. R. Fremantle, G.C.B., C.M.G. (Associate): My Lord Glasgow, Ladies, and Gentlemen, I feel some difficulty in commencing this discussion, and allowing it to range over the two papers, but I will endeavour to deal with them in the order in which they were read. With reference to the first paper, on Mercantile Auxiliaries, it is the opinion of the lecturer that they ought to be largely increased, and I think he gave very good reasons for that view. I am convinced that these cruisers with great carrying power would be very valuable—in fact, invaluable. It was, I think, at the discussion on Admiral FitzGerald's paper at our last meeting, that Commander Clover, the American Naval Attaché, stated, somewhat in opposition to the author, who had advocated scouts of 4,000 tons, that, in his opinion, the vessels which had been found most useful during their war with Spain, were those large mercantile cruisers such as the *St. Louis* and the *St. Paul.* I venture to think that both classes are wanted, but most certainly those large mercantile auxiliaries would be the vessels which would be used to go to distant parts of the world and carry information of the enemy's movements. It will be recollected that Nelson on one occasion, not long before Trafalgar, sent a vessel to the West Indies to give early information, and that vessel reached the West Indies in time. The same thing has occurred in very many cases in our naval history. It was by the despatch of the *Curieux* from the West Indies to Falmouth that Nelson sent the Admiralty news that the French fleet under Villeneuve was making its way back to Europe. Those are the sort of occasions on which these ships will be especially useful. Now, let us look simply at the question of the numbers that are required. We have now about 140 cruisers of various kinds. Nobody can look at the large area we have to control, and the enormous expansion there has been of our commerce, in every sea, without seeing that a very much larger number of cruisers is necessary now, than it was in former days. I quite agree with Lord Brassey that it is impossible for the number of war cruisers to be increased in proportion to the requirements for the protection of our Mercantile Marine. It is scarcely worth while to labour that point, but people who speak of the old war in 1812 to 1815, and the number of cruisers which they had then (some 500 of all sorts and sizes), scarcely realise that their case could be made a great deal stronger, if they would recollect that in those days, the area of warfare was practically restricted to the Mediterranean and the Atlantic, and that now, as we know very well, large fleets will be required in far distant China or Australasia. It was Nelson, who, speaking of battleships, said that "numbers only can annihilate." To paraphrase that, I say, certainly we must have numbers; but I see no way of getting those numbers except by multiplying mercantile auxiliaries. It is an unquestionable fact, as has been urged very often by Lord Brassey, that the Mercantile Marine must to a great extent be the nursery of our Navy. I hold that a Navy unsupported by an efficient Mercantile Marine is practically an exotic, and will only last a limited period. Now, on the question of subsidies, I quite agree that they ought to be granted liberally, and, after all, it is only a question of a comparatively small expenditure. £600,000 sounds a good deal, but £600,000 is only the price now of a second-class cruiser, or very little more, and certainly not the price of one of our newest first-class cruisers. The question of the protection of these vessels is a very serious matter, and one with regard to which I cannot quite agree with the author. I fail to see how any good protection could be given to the

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machinery of our large mercantile auxiliaries, at all events as their machinery is fitted now. Anybody who has travelled in them is aware that parts of the machinery are up to the upper deck, and the whole of their steering gear is very often entirely exposed, and frequently the steering engine, too, is on the upper deck. If we can be shown some way in which that can be protected, so much the better; but, irrespective of that, I would distinctly say we can make use of them. One of Lord George Hamilton's requirements was the absolute necessity for protection from the shot and highly explosive shell of modern quick-firing guns. I am not aware that in any of our ships that has been attempted. It may have been done in ships belonging to foreign countries, and, though I agree with the author that the protection of the men at the guns is the most important thing, I consider we want either a protective deck, or something in the way of protection for the machinery, if these mercantile auxiliaries are to be made anything like fighting vessels fit to meet even the smallest of regular cruisers. With reference to the conversion and re-armament of ships on the effective list, whatever is done should be done thoroughly and with some system. It is some years, I think, since Lord Charles Beresford read a paper in which he proposed that we should replace the muzzle-loading guns of the older ironclads, and put breech-loaders instead of them. I must say that I was entirely opposed to that proposal. Had it been carried out, it could not have made the ships efficient. The ships which are there referred to were, I think, older ships than those spoken of by Lord Brassey now, and the former had very small coal-carrying power. They were fitted in many cases with only single screws; they were masted ships, which would have required a great many alterations, and, even then, they would have been useless. But I have somewhat changed my opinion with reference to the alteration of ships now, and especially with regard to the more modern ships to which Lord Brassey has especially alluded. But, before stating my views on those subjects, I should like just to refer to a case within my own knowledge, of how, I venture to think, not to do it. H.M.S. *Téméraire* was built as a masted ship about 1878. She was a good ship in her day. While I was at Plymouth, the Admiralty ordered that she should be fitted with some quick-firing guns. They decided to put in some 4 in. guns, and somebody thought it would be a great waste to put good guns into her, so they got the oldest-fashioned quick-firing guns they could find, and put them into her. Of course the magazines had to be altered very considerably, and the question was, whether she was to retain her masts and sails, for it was generally understood that she was going to be a guardship somewhere, and the masts and rigging required a good deal of repair. I represented that it was not very useful to refit her aloft, so they did away with the masts and yards, and that again required that they should make it possible to hoist her boats in and out without masts and yards, which involved a question of derricks, and moving the bridge, and so on. In the case of the *Téméraire*, undoubtedly she was taken in hand haphazard, and a good many alterations were made in her, yet, after all, she was of no use. We had the same sort of thing in the *Hecate*, which was ordered to have new boilers. She is a useless ship, and I think the order has been cancelled; but it would have cost £30,000 to put those new boilers into her, and, if they had been, she would have been useless. I was very much struck with the remark of the author, with which I entirely agree, that the Admiralty has so much administrative work that a Consultative Committee would be useful in advising as to the reconstruction and repair. That I would press most strongly. Gentlemen, I have detained you quite long enough; but I should like to make some remarks as to the *Admiral* class, and the belted cruisers of the *Orlando* class. These two classes are in an entirely different position from those vessels to which I understand Lord Charles Beresford referred. The *Admiral* class, as Lord Brassey says, have some very good qualities indeed; but I am afraid they are too deep in the

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water. They have, however, good coal-carrying capacity; they have double screws, and are mastless ships. In many respects they are ships which I should think might be usefully converted. I find, taking from Brassey's book, that they vary in price from £724,000 for the *Collingwood* to £760,000 for others. The belted cruisers are in the same position; these vessels cost from £260,000 to £280,000; they are good steamers. True, they have got horizontal engines, which is rather a defect; but undoubtedly they might be made efficient ships (although that is a constructor's question), if a certain amount of expenditure was made upon them. I hope that some of the naval architects here will be able to give us some idea at all events as to what the expense would be. It is a question distinctly for a Consultative Committee, and what I want to point out, and lay special stress upon, is that these questions ought to be considered thoroughly, completely, and independently. It is a question of what the cost is, and what can be done. It would seem to me, in reference to the *Admiral* class, for instance, that we could possibly redistribute the armour and certainly replace the guns even with 9 in. guns of the latest pattern, 28 ton guns, and give some protection, which is so absolutely needed, to the batteries, and if that cost £200,000 per ship, it probably would be worth while to do it. I would also say, with regard to the *Orlando* class, that if we could carry out something like what is proposed by the author, and have longer ships—because they were always too short—if we could lengthen them and even spend £100,000 on each of them, that might also be worth while. Then there is the question of how long it would take; and that again is a question of detail, on which I am not competent to give an opinion. But I think there is a great deal in what the author proposes. I do think we ought, like the French and the Germans, to look into this question closely; and I am almost certain that, if we did, we should find it was worth while, and that it was necessary to do something in the way of reconstruction, such as has been proposed by the author. May I say one word in favour of that much-abused class in this country, the coast-defence vessels? Lord Brassey has referred to the *Hagen*, a coast-defence vessel of the *Siegfried* class. I venture to think those are very powerful vessels, and, although we do not, or hope we do not, require coast *defence* vessels, we may require coast *offence* vessels, and I would venture to remark, calling in aid a nursery simile, that what is most like a cat looking out of the window is a cat looking in. Gentlemen, I am afraid I have detained you at too great length, but these are very important subjects, and the two papers deserve full and adequate discussion.

Sir Nathaniel Barnaby, K.C.B. (Vice-President): My Lords, Ladies, and Gentlemen, it must have been very gratifying to Lord Brassey to have listened to the interesting contribution we have had from Admiral Fremantle on these two papers of his. I regret very much, for my part, that we have not had the presence here of Sir William White. He, especially with regard to this paper of the conversion and rearmament of ships, would have given us the benefit of his ripe, wise, and sound judgment. I, for myself, do not propose to say much about it, but I would suggest this, that Lord Brassey is not quite fair to the ships of the *Admiral* class, when he says that the secondary armament has no protection. It has heavily armoured bulkheads right across the ship, defending the guns in the secondary battery from right ahead and from right astern fire. I am sure that the two great main batteries, the armoured batteries, could be so dealt with as to find weight enough to enclose the guns of the secondary batteries within armour. I should now like to turn to the first paper read by Lord Brassey, which is to me an exceedingly interesting one. I entirely agree with him, as you will gather from what he has said in the paper, but he has omitted to say that he mentioned to me this matter, which was on his mind, before ever I read my paper in 1877. And in that paper, although he does not quote it, I say so. He also omits to tell you that it was he and Lord

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Northbrook who worked out, in conjunction with the Post-Office Department, the system by which the subsidies are now paid for such ships as we get. Now, how does this interest shipbuilders and marine engineers? It interests them here on the Clyde in this way: we all want rapid development; we want to get new things. We want, first of all, the building and designing of them, but we want that also because we wish this country always to have that which is newest and best in its naval armaments. Now the subsidising on a much larger scale than we have yet seen of ships built in advance of all other ships is good and necessary, and the reason is that there is no other way in which our great Navy can be kept constantly progressing, taking into itself the very newest elements. You cannot expect private shipbuilders and marine engineers to venture on the large changes which are indicated by Professor Biles, and by some others—changes such as are wanted in the liners in order to make them what they might easily be, efficient ships of war. But, gentlemen, you must pay money for it; and if you are prepared to do that, you will find men here on the Clyde, and on the Tyne, who are ready to give you all that you want. I am perfectly sure of that. It only needs that you should say, that the Admiralty must be told, that the men who are concerned in the designing of the ships, and in the designing of the engines, are the men who can put into the Royal Navy constantly new elements, strengthening it by every day's improvements. For you observe that the ships which are so built do not become obsolete in the same sense as the cruisers for which I was responsible, and for which Sir William White is responsible. These become obsolete very quickly, because the main element in a cruiser is speed, and as soon as she begins to fall off in speed, the naval officer begins to throw her out of his calculations. But the shipowner does not do that; the shipowner has always got something useful for a ship to do if she is a first-class vessel. If he cannot find it, somebody else will. You have been told by Lord Brassey that the *Magenta*, a ship in the French Navy, only ten years old, and costing a million of money, has already had to be transformed. I say nothing of the ships with which I am connected. These are so old, and I am so old, that I am quite prepared to hear them condemned as obsolete; but I observe that some of the ships designed by Sir William White—battleships, and not cruisers—are spoken of now as "obsolete." I think that the gentlemen who write on the subject forget that, while a cruiser can become obsolete because speed is its main quality, it is not the same with a battleship. Speed is not here the main quality; there are other qualities, and I have often thought that if a well-built and well-protected ship is kept up by attention, and by putting in such improvements as can be made without much expense, she will, even at the end of twenty years and longer, be worth at least half what she was when she was built. I am speaking now only of battleships, and not of cruisers. Directly the element of high speed comes in, all is altered, and you cannot do anything; but in the other ships, I do not believe they lose in value more than about 2½ per cent. per annum from the time when they were first built. I think that at the end of twenty years, 50 per cent. has gone off their value; but, in my judgment, the sailors would find those ships are worth, even at the end of that long period, at least half of what they cost to build. Lord Brassey refers to what has been done by the Germans. Our German friends here, whom we are very pleased to see, have been distinguished by him as having gone on right lines. I think they are on the right lines. They are building these large ships, which will always form most efficient auxiliaries to their Navy, and they will do what appears to me to be necessary in Germany, something to get that economy which is desirable in order to make things go smoothly with their Parliament. There are those who are constantly wanting a strong Navy, and there are those who do not want to spend the money. We should not mind spending the money, if we were able by that expenditure to carry out such an excellent

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proposal. Lord Brassey has told us, comparing what we are spending on new ships and engines for this year with that which we spent in the years 1872-73, when we thought we had a powerful Navy, that we are spending in this year eleven times as much for those new constructions as we did in 1872-73. Our German friends will make a note of that, and will see why Lord Brassey commends what they are doing. It may be difficult for us in England to get the proposition, which Lord Brassey has put before us, carried out. He proposes here something with which I entirely concur. He says: "Taking into consideration the fact that our supremacy depends on the efficiency of our Naval and Mercantile Marine, a committee of Admiralty officials, shipowners, and shipbuilders should be formed to discuss the best method of constructing a combined Naval and Mercantile Marine." My Lords and Gentlemen, I venture to hope that much good may follow from what Lord Brassey has done. He is the best-known man in this room—a man who is known to sailors all over the world. If due weight is given to what he has proposed, and such a committee as that is appointed, this Glasgow Exhibition and this meeting will become famous in history.

Professor J. H. Biles (Member of Council): My Lords, Ladies, and Gentlemen; Lord Brassey, in the able paper which he has given us on Mercantile Auxiliaries, has perhaps not emphasised so much the question of the practicability of making these mercantile auxiliaries capable of becoming warships, as has been done on previous occasions; and Admiral Fremantle must have overlooked the fact that in 1895 I read a paper before this Institution, in which I attempted to deal in considerable detail with the steps which it would be necessary to take, in the preliminary stages of the design of a mercantile steamer, in order that she should be of great use in war time. Lord Brassey was good enough to refer to a paper which I read at the Engineering Congress of the Civil Engineers, but that paper was a mere *résumé* of the former one, which, while it stands undisputed to-day by any authority known to me, in its facts and its figures, has, I regret to say, received practically no attention, until the subject has had to be resuscitated by Lord Brassey to-day. Lord Brassey probably remembers the speech which he was good enough to make upon that paper, and, had it not been that national duties took him away from this country, it is probable that long before this, we should have heard the powerful appeal that has been made to us to-day. The question that has been brought up here is, in the first place, a political one, dealing mainly with the naval policy of the Government. It is also an economic question, dealing as it does with the best method of spending money for Imperial defence; but when the naval policy question is decided—and with it the economic one—there must come finally to this Institution, or to some of its members, this further question to be settled: Is it practicable to make mercantile steamers so that they can be of use in war time: not only as scouts; not merely, as Admiral Fremantle says, to run here and there to find out pieces of information; but whether they can be made capable of standing in the line of battle in the same way that some of our first-class cruisers are capable of? And, also, what would be the cost, and sacrifice of essentials, if such a project were carried out? Perhaps I may be forgiven if I recall the results of the calculations that were arrived at in my paper dealing with this question. I there attempted to show that a first-class mail steamer can be adapted in her original design so that she could in war time be made little, if at all, inferior to our first-class cruisers in gun, torpedo, and protective capabilities; while in speed and coal endurance she would be superior. I took the case of a ship of the size of the *Etruria*, and I showed that for 4 per cent. of her displacement appropriated to structure in the original design, we should then have a vessel which would be capable of carrying side armour for three-fifths of her length, and of bearing an armament equal to that of a first-class cruiser of that date. Now, of course, this increase of

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displacement, which means an increase of weight of structure, is something which detracts from the earning power of the ship, and the question is, what is the cost to the shipowner? What would the Government have to pay, in order that the ship should be able to return to the shipowner at least the same profit that she would if she were not adapted for war purposes? I have dealt with that in detail, and have not yet seen any dispute as to the accuracy of the results obtained. Therefore, while it stands to-day unchallenged, perhaps it may be of interest to restate the figures given, in this discussion. An annual expenditure of £19,000 to the shipowners represented the interest and depreciation, and the alterations, or rather the modifications, in the design of the ship, which would be required to make her equal to a first-class cruiser; but, of course, in addition to that, there would be the cost of guns and armour which would have to be made and fitted to the ship, taken off and put in the store, while the ship was carrying on her peaceful duties. All ships must have a hull, and all steamers must have both a hull and machinery. The difference between the mail steamer and the warship is, that one carries cargo and passengers, and the other carries armour and guns; and if we can adapt mail steamers so that they can carry cargo and passengers in time of peace, and armour, and guns in time of war, then we shall have added something to the fighting strength of the country. In war time, the ship puts on her armour, takes down her guns, and with the muscles and sinews which have been well trained in peace, stalks forth a man-of-war in search of her country's foes. She represents the most modern idea of the soldier—the useful, hard-working citizen in time of peace, who, when war time comes, goes forth to fight with as good a spirit as the professional soldier, and not infrequently with as great and glorious a result. The question Lord Brassey puts (and he put it many years ago, before the Boer War had shown what our citizen soldiers can do) is: Have the resources of our non-fighting fleet been sufficiently considered in our scheme of national defence? Have our mercantile auxiliaries been starved, as our Volunteers have been? We may now ask the question, Will it be necessary to have a disaster on the sea in order to bring home to our statesmen the value of our citizen ship? Shall we be told by them that we know no more about the value of mercantile auxiliaries than the man in the street? I think this Institution, which is not a "man in the street" on this subject, will be well advised if it takes such action as will support Lord Brassey in his endeavour to awaken the authorities to a full sense of the value of mercantile auxiliaries, and encourage their owners to design them to be of real fighting use in war time.

Admiral Fremantle: Might I ask what was the size of the ship on which you made that calculation?

Professor Biles: A ship of about 14,000 tons displacement—a mail steamer, the *Umbria* or the *Etruria*—you will probably know those steamers—500 ft. long and 57 ft. beam—the fair average size of the larger mail steamers that are being built to-day in comparatively considerable numbers, compared with what was being done some years ago. I am afraid that I have already taken up too much time to deal with the questions raised in the other paper. One of the unfortunate things that falls to the lot of this Institution is that we have more papers than we have time to read and discuss. It is very unfortunate when the papers are of the calibre of this one, and are represented by such readers as Lord Brassey. Admiral Fremantle reminded us of a very important thing in connection with the conversion of these older ships: it was that the *Téméraire*, after being converted in a haphazard way, as he put it, was then no better than before she was converted. I think, if that is the case, we certainly should associate with the proposal to convert any of the older ships, the proposal of



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Lord Brassey: that a committee of some kind should be appointed, which should prevent the conversion of these ships in such a way that, when converted, they are no better than before.

Admiral Sir NATHANIEL BOWDEN-SMITH, K.C.B. (Associate): My Lord President, Ladies, and Gentlemen, I am not ready to make any remarks worthy of your consideration upon the two papers which we have just heard read, because I only had them put into my hand as I came into the room, and, not being a very ready thinker, I like to put things under my pillow and sleep on them before expressing an opinion. As to going into the figures which Lord Brassey has placed before us, that would be impossible. It is for the Naval Architects to decide how far auxiliary cruisers can be made available for fighting purposes, and Professor Biles, who has just spoken, has given us his views on that subject. With reference to the comparisons between ships of our Mercantile Marine and those of foreign Powers, we must acknowledge that they are very unpalatable facts, and not pleasant reading; but I take comfort in the fact that, although some of the ships of our Mercantile Marine may be inferior in speed, yet as regards numbers we still have a very great superiority, and I remember how at the present moment, without dislocating our trade in the least degree, and having all our mail steamers running their accustomed routes, we have been able to transport over a quarter of a million of men to South Africa, together with all the necessary horses and stores, and, to the best of my knowledge, entirely under the British flag. With regard to the mercantile cruisers which the first paper dealt with, I think the first consideration should be to endeavour to maintain our large, fast steamers running their accustomed routes, and also to keep open the trade routes of those carrying food to the British Isles. Those which have an armament prepared, might carry their guns for self-defence, but as to how far they would be justified in continuing an engagement should they come across another auxiliary cruiser, must rest entirely with the captain of the ship. It would not be right to endanger the lives of the passengers, or run any undue amount of risk. It appears to me that one excellent reason for retaining, by a subsidy, the services of our fastest steamers is to prevent them, in an emergency, being leased or sold to any other Power. Before reading his papers, Lord Brassey said the time had nearly arrived when he ought to be coiling up his ropes and lying by. I can only say that I voice the opinion of my brother officers and, I hope, of the members of this Institution, when I say that I trust it will be a long time before he does coil up his ropes, and before the country loses his valuable assistance in developing not only the Navy, but our great Mercantile Marine.

Mr. FRANCIS ELGAR, LL.D., F.R.S. (Vice-President): My Lord, Ladies and Gentlemen, after the long and interesting discussion we have had, I shall confine my remarks to the point which, I think, is the real practical outcome of both of Lord Brassey's papers. There cannot be a doubt in the minds of shipbuilders, that everything Lord Brassey suggests in his two papers can be done: it is all perfectly practicable. But it is of course very much a question of money, and especially whether money applied to increasing the efficiency of the matériel of the Navy could be better spent upon mercantile auxiliaries, and in bringing old battleships up to date, than in any other way. I agree with Professor Biles that there would be no practical difficulty, nor much expense involved, in making our fast mercantile liners as efficient in the way of gun protection as the present cruisers, while preserving higher cruising speeds and greater coal endurance. The greatest difficulties would arise in attempting to give effective protection to the machinery and boilers; and the cost of adapting a mercantile liner for service as an auxiliary cruiser would depend chiefly upon the extent to which it might be considered desirable to go in that direction. Now, the reason I venture to intervene at

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all in this discussion is that Lord Brassey in his paper, and Sir Edmund Fremantle in his remarks, ask for the opinions of shipbuilders, not merely upon the technical merits of these proposals, but more particularly as to what it would cost to carry them out. Now, my Lord, I venture to say that no shipbuilder could give an approximate estimate of cost, right off, that would be in any degree trustworthy, even if he knew exactly what alterations of the vessels as now built would be required, and what work would be involved. But we do not know yet what alteration would be required, or what standard of efficiency as an armed cruiser would be adopted by the naval authorities for mercantile auxiliaries. It would also be impossible to give any idea of what it would cost to bring the ships named in Lord Brassey's second paper up to date, without full details of how it is proposed to treat them. The standard of efficiency to which they might with advantage be brought would depend, as in the case of mercantile auxiliaries, upon the economical question of whether the various improvements which might be practically or mechanically possible would be worth the money they would cost. What I desire to urge upon this meeting is that we should strongly support the proposal made by Lord Brassey and by the other speakers, that the whole question should be referred to a consultative board. It is not a question that could possibly be settled by individual opinions; but it is essentially one for a board upon which would be represented the naval officers who would require to use the ships, and who would be able to say what was wanted of them; the Admiralty contractors; and also private shipbuilders, whose opinion might be required. I do not know any other way of obtaining an answer, that would be worth anything, to the question of the cost involved in these proposals; but I feel sure that such a Board or Committee, if properly constituted, would arrive at a definite result, and would answer, once for all, in a manner that would command public confidence, the two very important questions raised by Lord Brassey in his paper, as to the most effective way of utilising the vast fleets of our Mercantile Marine in case of war, and as to the best way of dealing with battleships that are becoming obsolete. I believe, with Sir Nathaniel Barnaby, that if, as the result of this meeting, some step of the kind proposed were to be taken, we shall have done good service to His Majesty's Navy, and made this year memorable in the history of our Institution. There is one thing I would like to add with reference to the paper upon the "Conversions and Re-Armament of Ships," and it is that I am sure Lord Brassey's estimates of the cost of making the alterations suggested, and of the time requisite for carrying them out, are very sanguine.

Admiral the Right Hon. Sir JOHN DALRYMPLE-HAY, Bart., K.C.B., D.C.L., F.R.S. (Vice-President): My Lord, I do not wish to detain the meeting many minutes, but I would like to come to a practical conclusion after these two very valuable papers which have been presented to us by one of our late Presidents, and it seems to me that I might ask the meeting to urge upon our President to approach the authorities—the Admiralty—and present a petition agreed to by this meeting in terms to be modified, if necessary, but embodying the suggestion made by Lord Brassey. I find in his paper there is a paragraph which might form the subject of the petition to the Board of Admiralty from this Institution, urging that our naval supremacy depends upon the efficiency of our Naval and Mercantile Marine, and that a committee of Admiralty officials and shipbuilders should be formed to discuss the best method of constructing a combined Naval and Mercantile Marine. If anybody here will second it, I will put forward that proposition, to be modified by the Council under the direction of the President, to embody the suggestion made by Lord Brassey in his first paper. This would perhaps induce the authorities to move in the direction which we all recognise would be of great benefit to the country.

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Mr. J. I. Thornycroft, LL.D., F.R.S. (Vice-President): My Lord, I should like to second this proposal, which I really think has the support and best wishes of all of us here. I know in Glasgow they wish to economise money and get the most they can for it, and I think this proposition may aid in that direction.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Perhaps Sir John Hay will allow this proposition to be postponed until after the next paper is read.

The Right Hon. Lord Brassey, K.C.B., D.C.L. (Past President): I am told it is usual for the readers of papers to reply. I can only say that I am exceedingly indebted to those that have spoken, for the valuable contributions they have made to the discussion. I entirely support the proposal which has been made by Sir John Hay, and seconded by Mr. Thornycroft, that we should approach the Admiralty in reference to the appointment of a committee.

The following communication relative to the paper under discussion was received from Admiral Sir J. O. Hopkins, G.C.B. (Associate): I hold strong views both on the "Conversions and Re-armament of Ships on the effective list," and also on the subject of "Mercantile Auxiliaries." In regard to the first, I may mention that in a paper I read at the United Service Institution in December last, I advocated, *inter alia*, the *Admiral* class being reconstructed, unless replaced, but, as the latter seems at present out of the question, I am strongly in favour of "conversion." I am also strongly in favour of putting the work out to contract, as the Royal Dockyards are too full of other work to attempt more at present. I may be pardoned for quoting from the paper I read, a rather fuller extract than the one bearing on the conversion of the *Admiral* class, but as part of a reconstruction scheme, I insert it:—"We may also hope early in the coming century to get rid of all muzzle-loading guns by eliminating from the Navy List all ships carrying them. By this measure, the simplification of our ordnance and all its accessories would alone be an untold blessing, as then, roughly speaking, we should have only nine descriptions of heavy guns, whilst we should get rid of about a dozen kinds of obsolete weapons for which powder, projectiles, stores, &c., must be maintained, as also spare guns and spare mountings, not to mention the occupation of most valuable storehouse room by these obsolete weapons. There are two ways of carrying this out: one being to let time work out the change, as is apparently the present intention; the other, to face the situation, ear-mark the ships to drop out, and commence the replacement at once. For the sake of illustration we will take haphazard the following ships with muzzle-loading guns, none of which vessels, by their peculiar construction, can mount an auxiliary armament of breech-loaders, so are practically useless against modern ironclads. The ships are:—*Agamemnon, Ajax, Neptune, Superb, Téméraire, Inflexible*, whose united displacement is 56,620 tons. Working to this tonnage will give us four 14,000-ton ships, or five of 11,000 tons. Let us assume we prefer the five, and construct them as improved *Barfleurs*, two of them to be started at once, two the following year, and one the year after, on a supplementary estimate, so as not to cut across the ordinary shipbuilding programme. Accepting the time for construction as three years, we have *Téméraire* and *Neptune* dropping out in 1903, *Superb* and *Inflexible* in 1904, and *Ajax* and *Agamemnon* in 1905, with the remainder to be dealt with then on similar lines. Let us also face the replacement of cruisers, or eliminate at once from the Navy List all obsolete first-class cruisers of the *Nelson* type, as also any other now called first-class cruisers, such as *Agincourt, Northumberland*, and *Minotaur*. Then, early in the next century, may we not look forward to the *Admiral* class being replaced, or, failing that, reconstructed with a view of so reducing

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present weights as to provide plating, or casemates, for their 6 in. auxiliary armament? I cannot help thinking that their present 18 in. belt reduced to 6 in., as in the *Glory*, their heavy barbette guns changed to 10 in. guns, as in *Renown*, and other weights carefully cut down, the alteration can be effected. Let us also contemplate the change of guns from muzzle to breech-loaders in such ships as *Monarch* and *Dreadnought* if still retained on the effective list." In respect to the other question of "Mercantile Auxiliaries," I am entirely in accord with Lord Brassey's views, and can urge nothing stronger or more to the point than his paper presents; in fact, I may say that both papers represent strongly the views of many Naval officers, and, if acted upon by the authorities, they cannot fail to increase our fighting strength afloat in a marked degree.

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# ON THE LIMITS OF ECONOMICAL SPEED OF SHIPS.

By E. H. Tennyson-d'Eyncourt, Esq., Member.

[Read at the Summer Meetings of the Forty-Second Session of the Institution of Naval Architects, June 25, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair].

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A wish has frequently been expressed at meetings of this Institution that members would give results of their experience derived from the actual performances of vessels on progressive trials, and the methods they adopt in estimating the power necessary for driving ships at given speeds: more especially was this the case when Mr. James Hamilton read his paper on "Horse-Power deduced from Progressive Trials," some three years ago.

I have, therefore, endeavoured to bring into line the data obtained from a large number of progressive trials, of ships of different forms, and to derive certain broad practical rules from these data, which may, I trust, be of interest, and may possibly elicit the results of their experience from others who have made a study of the question.

I think it has, perhaps, been too common a practice in fixing the speed for a vessel, and in estimating the necessary indicated horse-power for that speed, to overlook the question of what speed is appropriate to the form of vessel under consideration, and to go upon the broad basis of allowing plenty of margin of horse-power, with the result that often too high a speed has been aimed at, and too much power has been allowed. As a necessary consequence the weight and cost of the excess has been a continual burden upon the earning capacity of the ship. This was, perhaps, of no great consequence when competition was not so keen, but nowadays, when cost has to be cut down to a minimum in every direction, not only by the shipbuilder, but also by the shipowner, the question of giving a ship enough, and not too much, power and speed has become one of first importance.

It is necessary to bear in mind the fact that every ship has an appropriate limit of speed, beyond which it is most uneconomical to attempt to drive her; or, in other words, any increase of speed beyond this limit requires an undue increase of power. It has,

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therefore, been my object to arrive at a method of defining this limiting speed for ships of different sizes and different forms.

To attempt to arrive at a general solution to meet all cases is, I need hardly say, an impossibility; but I venture to think that a fair approximation to the suitable power and speed may be arrived at, by a consideration of the general principles governing the relation between the dimensions and form of ships, on the one hand, and the appropriate speed and power, on the other.

To obtain accurate values, it is, of course, necessary to consider each case individually upon its own merits; to have the exact form of the under-water body, and a record of the performance of a similar ship, or of model experiments.

In analysing the trial results which I have had at my disposal, I have not attempted to divide the indicated horse-power absorbed into all the heads, which, in making an exhaustive analysis, would, perhaps, be necessary; but I have, for the sake of ease in comparing the performances of different vessels, always assumed the effective horse-power to be half the indicated horse-power. This is a somwhat crude method, perhaps, but I think as good as any other, where so many unknown (or at least doubtful) quantities are involved. I think, also, this is a fair average value to assume for the propulsive co-efficient, with modern triple or quadruple expansion machinery and suitable propellers, at the higher speeds which I am considering.

Having, therefore, made this assumption, I have proceeded to calculate the horse-power absorbed by skin friction on the basis given by the late Mr. Froude, and now universally adopted; and the difference, after deducting the skin horse-power from the effective horse-power at any speed, gives the power necessary to overcome the so-called residuary resistance, which at the higher speeds is principally caused by wave-making. For brevity's sake this may be termed the wave horse-power.

The different values of the wave horse-power obtained by this method for similar ships, and in some cases for sister ships, show up the variations in the value of the propulsive co-efficient, of propeller efficiency, &c., actually occurring in practice; but the merit of the method lies in the fact that it gives a definite comparison between the performances of different ships. The naval architect must make allowance for the efficiency of machinery, &c., as experience dictates.

Having obtained curves in the manner indicated for many ships of varying dimensions, forms, and degrees of fineness, I have been able to deduce the general results shown in the following table. This gives the co-efficient of fineness and the amount of parallel body which may be associated with that co-efficient, and the corresponding limiting economical speed, expressed as a percentage of the square root of the length of the ship in feet on the water-line.

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| Co-efficient of Fineness. | Parallel Body as Percentage of Total Length of Ship on W.L. | Limiting Economical Speed in Knots as Percentage of $\sqrt{L}$ in Feet. |
|---|---|---|
| ·5 | 0 | 100 |
| ·6 | 14 | 86 |
| ·7 | 30 | 68 |
| ·8 | 49 | 48 |

Diagram I. (Plate XLIX.) shows these values in curves P P and L L. The former gives the appropriate length of parallel body, which is equivalent to giving the combined lengths of entrance and run, and it will be seen that the percentage values of $\sqrt{L}$ for the limiting economical speed are practically the same as the percentage values of the combined length of entrance and run. This agrees with the value which has been given previously for vessels with no paralled body, but gives a smaller economical speed for vessels with parallel body; the previous rule having been that the economical speed is equal to the square root of the sum of the lengths of entrance and run, and it seems to show that a penalty must be paid for introducing parallel body.

Taking actual trial results, I find that the indicated horse-power at the limiting speed, as defined above, is varying as the fourth power of the speed, and varies, in increasing ratio, till at about 12 per cent. above the limiting speed it is varying as the seventh power of the speed; whilst the wave horse-power varies as $v^7$ at the limiting speed, and as $v^{10}$, or sometimes as a higher power of $v$, at 12 per cent. above the limiting speed; and that at this point, viz., about 12 per cent. above the limiting speed, the wave horse-power is approximately equal to the skin horse-power, and then rises above it, the skin horse-power, of course, always varying as $v^{2\cdot83}$.

Diagram III. (Plate L.) gives a typical curve of horse-power, A A being the curve of effective horse-power, with 100 as its value at the limiting or unit speed. Curve B B gives the skin horse-power, and curve C C the wave horse-power. I have not taken the curves below ·6 of the limiting speed, as it is not necessary to do so for my present purpose.

The ratio of wave horse-power to skin horse-power at the limiting speed depends, of course, upon the form of entrance and run and the mean girth of the vessel up to the water line; but ½ seems to be a fair average value for vessels of fine entrance and run and full midship section. If the midship section is fined, and the wetted surface thereby reduced for the same block co-efficient, it naturally follows that the entrance, or run, or both, must be filled out, with the effect that the skin resistance is reduced, and the wave-making increased, bringing the curves of skin horse-power and wave horse-power more closely together, as indicated by the dotted

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curves on the diagram, and bringing the point where skin horse-power is equal to wave horse-power to a lower speed.

It is, therefore, impossible to lay down general rules to meet all cases, as the variations which may occur are almost infinite in number, and so only an indication of the kind of thing that may be expected can be given. Considerations of stability, involving beam, or fulness of water-line, or of girth, which affects weight of hull, more especially if the vessel is to be built to Lloyd's rules, or the question of docking facilities, are merely examples of what the designer has to take account of, and they often hamper him in adopting what he knows would be a better form, or more suitable dimensions for the speed than those he is compelled to take.

The results I have obtained are from the trials of vessels of good form for the co-efficients of fineness they possess. But there are many ships whose performances are not so good as I have indicated, and there are others again with better performances; the latter, however, generally speaking, appear to have had rather better propulsive co-efficients than the 50 per cent. I have assumed; one or two examples apparently having values of 57 to 58 per cent.

The values of the Admiralty constant,

$$C = \frac{D^{\frac{2}{3}} v^3}{\text{I. H. P.}},$$

where

D = Displacement in tons,
$v$ = Speed in knots per hour,

at the limiting speed, as found above, are fairly uniform, varying, however, with the size of ship as given by the table below :—

| Length of Ship on Water Line. | Value of C at Limiting Speed. | Value of C when Skin H.P. = Wave H.P. |
|---|---|---|
| 200 ft. | 225 | 188 |
| 300 ft. | 256 | 207 |
| 400 ft. | 278 | 224 |
| 500 ft. | 295 | 237 |
| 600 ft. | 310 | 246 |

Diagram II. (Plate XLIX.) shows these values of C plotted on a base representing length of ship.



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These values of C are less for the smaller ships, on account of the higher coefficient of friction for the shorter length of vessel, and also on account of the error in the assumption of the general truth of the law of comparison when applied to the whole resistance. These two causes are not, however, sufficient of themselves to account for the decrease in the value of C. I think this decrease is partly due to eddy-making, caused by minor projections and irregularities in the surface of the smaller ships, these irregularities being on a proportionately larger scale in the smaller than in the larger vessel. It must also be borne in mind that any roughness of sea or wind has a greater effect in proportion on the smaller ships, tending to spoil their performances on trial, as compared with those of larger ships.

I do not intend, when I use the term "economical" speed, to make it include the financial side of the question; as it frequently happens that it is cheaper to increase horse-power, to get a little extra speed when required, than to increase the length of the ship, or to fine her lines. What I do intend to convey is that, if a speed be aimed at, slightly above the limiting speed I have defined, with the indicated horse-power increasing as more than the fourth power of the speed, it will be found possible to drive at that speed, with the same horse-power, a vessel increased in one dimension, viz., in length, and proportionally increased in displacement. Whilst, if a still higher speed be aimed at, the vessel may be increased in two dimensions, viz., length and draught, or length and beam, and still attain the higher speed, with the same horse-power as the smaller ship. Finally, as was proved by the late Mr. Froude, when the indicated horse-power is increasing as the seventh power of the speed, it is possible to drive a similar ship, increased in all three dimensions, at the same speed, with the same horse-power as the smaller vessel, and this is more than borne out by the facts.

I have shown that the indicated horse-power is usually increasing at this rate, viz., as $v^7$, when skin horse-power = wave horse-power, or at about 12 per cent. above the limiting speed. This is then the speed at which it becomes economical to increase all the dimensions of the ship in the same proportion; of course, from the point of view of driving the greatest displacement at the given speed.

There are many vessels in which this limiting speed is far exceeded, such as fast cross-Channel passenger vessels, torpedo gunboats and destroyers; but these types are for very special requirements, and are not economical in the sense of carrying large displacement at the required speed. They are not, therefore, included in the scope of this paper. Nor do the figures I have given apply to vessels of very great beam in proportion to their length. The examples I have taken are all of ordinary proportions, not less than $6\frac{1}{2}$ beams in length, and of ordinary draughts; but, as this includes practically all mercantile vessels, and also cruisers of the usual dimensions, the range is fairly wide.

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In these two latter classes the number of indicated horse-power per ton of displacement rarely exceeds two. In the slower vessels of the Mercantile Marine it goes as low as ¼ I.H.P. per ton of displacement. In the torpedo-boat destroyers it goes as high as twenty, and in H.M.S. *Viper*, with the turbine machinery, has, I understand, reached about thirty. So that the exclusion of these very fast craft from a paper dealing with the economical side of the speed and horse-power problem is obviously reasonable. The very broad vessels, also to some extent abnormal, have, therefore been excluded, as have also the performances of paddle boats.

As regards the comparative performances of single and twin screw vessels there appears to be little difference in vessels of good form.

Reverting to the limiting economical speed as defined, I find that, generally, for 1 per cent. increase above this limit of speed, 5 per cent. increase of horse-power is necessary; and at the speed when skin horse-power = wave horse-power, or about 12 per cent. above the limiting speed, 1 per cent. increase of speed requires 10 per cent. increase of power. Taking these figures, it appears to me that any increase of speed above the limits indicated should be most carefully weighed before adopting it, in view of the disproportionate increase in power and weight of machinery thereby rendered necessary, together with increased engine-room staff, the larger coal bill, and the reduced weight and capacity for cargo. I think there are ships that might well have been made finer, or have had their speed and power reduced when in the design stage, at the cost of foregoing that last half knot, which possibly requires from 20 to 25 per cent. additional horse-power, enabling the vessel, perhaps, to steam a distance of 12 knots farther in twenty-four hours at the cost of burning out coal which would keep her a whole extra day at sea at half a knot less speed, and this without taking into account the original saving in weight and cost of machinery which might have been made.

In conclusion, I would say that I have had great difficulty in obtaining reliable data of progressive trials of ships with very large co-efficients of fineness. In these vessels, usually, no progressive trials are made, and, in any trial that does take place, the propellers are frequently only partially submerged.

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## DISCUSSION.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, we have had a very interesting paper from Mr. d'Eyncourt; but, before proceeding to the discussion of that paper, I will ask Sir John Dalrymple-Hay if he will be so kind as to read out to us the motion which he gave us briefly just now.

Admiral Sir JOHN DALRYMPLE-HAY: My Lord, I will not now again detain the meeting by stating the reasons for putting forward this motion, as I have already given them on a former

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occasion. The form of the motion will be as follows: "As our supremacy depends on the efficiency of our Naval and Mercantile Marine, a committee of Admiralty officials, shipowners, and shipbuilders should be formed to discuss the best method of constructing a combined Naval and Mercantile Marine." Those are the words taken from the paper which Lord Brassey read, and they are in the form which I believe the seconder approves of.

Mr. Thornycroft: Yes.

The President: I believe Mr. Thornycroft seconds that motion.

Mr. Thornycroft: With pleasure.

The President: If nobody has any opposition to this motion, I will, as time is short, put it to the meeting; but before doing so, I think I may say that we are all extremely grateful to Sir John Dalrymple-Hay for having brought forward this suggestion. It is one which apparently falls in with the ideas of all present, and I anticipate it will be carried unanimously.

The motion was then put to the meeting, and was carried unanimously.

Mr. Francis Elgar, LL.D., F.R.S. (Vice-President): My Lord, Ladies, and Gentlemen, I consider we are indebted to Mr. d'Eyncourt for a very interesting paper, on an important practical question connected with the designing of ships. It will be remembered by some who are present here to-day, that in the early years of this Institution it was believed by many that there was an absolute limit to the speed of a ship; and that this limit, and the speed which it is possible to reach in any vessel, was absolutely dependent upon the square root of her length. It was then generally supposed that, no matter what power might be put into a ship of a given length, it would be impossible to drive her beyond a certain definite limit of speed. We have, happily, advanced in knowledge upon such points since that time. Mr. Thornycroft exploded the theory of limited speed by publishing the report of the trials of his fast little river boat, the *Miranda*, which was the forerunner of the present torpedo boats and torpedo-boat destroyers. We know now that what was once thought to be the limit of a ship's speed is represented upon a curve of resistance merely by an irregularity of curvature, caused by the resistance increasing at an abnormally rapid rate over a comparatively short range of speed; and that this abnormal rate of increase of resistance is due to the interference of the bow and stern waves at that particular speed. It is the portion of the curve of resistance at which this state of things occurs that Mr. d'Eyncourt deals with in his paper; and he has given what appears to be a very fair approximate rule—as embodied in his tables and diagrams, which are based upon experience gained by the trials of ordinary types of ships—for indicating the speed at which that critical portion of the curve of resistance is being reached. His results are limited to the trials of steamers of ordinary types and proportions. It is known of course that the rate of increase of resistance, at the portion of the curve of resistance under consideration, varies in vessels of different proportions. In vessels which are very broad and very shallow in draught of water, the rate of increase is low. In vessels that are narrow and of very deep draught, it increases with great rapidity. In our torpedo-boat destroyers, we have an example of boats of comparatively great length in proportion to their displacement. They have a displacement of only about 300 tons, with a length of about 200 ft.; and, in their case, the increase of resistance at the critical speeds is merely represented by a flattening of the curve of resistance. The rate of increase of resistance at that point seems to depend more upon the relation of the displacement to the length of the ship, than upon other features of the design. The question of whether what Mr. d'Eyncourt terms the "limit of economical speed" is being exceeded in the

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design of a ship, is sometimes of the greatest importance in estimating the speed, and is a fruitful source of error in speed estimates. It is one of those points which it is impossible to investigate thoroughly in the absence of speed trial data of similar ships, without recourse to model experiments. Those who have no means of making model experiments are much indebted to Mr. d'Eyncourt for putting before them the results he has been collecting in the course of his experience, results that bear upon this very important practical point. I would like to point out, with reference to the help that might be obtained in solving difficult speed problems if recourse could occasionally be had by shipbuilders generally to model experiments, that we are in an unfavourable position in that respect in this country as compared with other countries. I was very much struck during a visit I paid to the United States a short time ago, by the extent to which mercantile shipbuilders in that great country are assisted, in dealing with difficult questions of speed, by the Navy Department at Washington. I then had the privilege of visiting the experimental tank at Washington—which is doing admirable work both for the Navy and Mercantile Marine under the superintendence of Mr. Taylor, Naval Constructor at Washington, one of our most valued members—and I was surprised to find that Mr. Taylor was not only making experiments with models of ships of the Navy for the information of the Construction Department, but also with models of mercantile ships for the benefit of private shipbuilders who wished to have the best speed data procurable for the designs they had in hand. It was very interesting and suggestive to me, never having heard of such a thing among ourselves, to see models of mercantile steamers that had been sent by different private yards of the country, going through their trials in a very expeditious manner. I am sure the United States Naval Department is acting wisely and truly in the public interest, by thus fostering the growing industry of shipbuilding, and giving mercantile shipbuilders the benefit of their own superior scientific knowledge and appliances. I asked Mr. Taylor what the Navy Department charged for outside work of that kind, and he told me that, for making a model from the drawings furnished by a shipbuilder, conducting the experiments, and supplying the shipbuilder with a set of resistance curves, they merely charged the out-of-pocket expenses upon materials and wages. I understand from the shipbuilders themselves that an ordinary charge is about £50. I am sure it would be an inestimable advantage to private shipbuilders in this country, to have the means of getting model experiments made, in cases to which the trial data of existing ships do not apply with sufficient exactness, or where unusually difficult questions of speed have to be dealt with.

Mr. James Hamilton (Member of Council): My Lord and Gentlemen, it is only just now that a copy of Mr. d'Eyncourt's paper has come into my hands. I can say at once, however, that I agree with a good many of the conclusions that I see in this paper. It is very clearly written, but at the same time it is a very complicated subject, and for anyone to venture to discuss this paper without previous investigation would, I think, be a poor compliment to the author. I have scarcely realised what the limit of speed is: I see a curve, and I see it varies in different percentages, and very properly so, no doubt, but it is very difficult to say much about the question without very careful study. It is apparently an extension of the paper that I read two years ago, when I dealt with the horse-power at speeds equal to the square root of the length. The writer of this paper has gone rather further, it seems to me, and has found a limiting speed for each form and each size of ship. I will not take up any more time, but I would just like to say, and I think I speak the feelings of every member of this Institution, that we are glad and proud to see a young member—I believe Mr. d'Eyncourt is one of our youngest members—reading such an admirable paper as this.

Mr. Archibald Denny (Member of Council): My Lord and Gentlemen, I am very much in the same

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position as Mr. Hamilton. I did get the paper at the end of last week; but, as I received the other papers at the same time, and endeavoured to read them all, it must be evident that I have not had much time to consider it. I would like to say, however, that Mr. d'Eyncourt quite sees his difficulty with regard to assuming a constant efficiency, and he knows as well as I do, that the efficiency does not remain at 50 per cent. We have examples from our experimental tank of efficiencies quite as low as 35 per cent., and, on the other hand, we have them as high as 65 per cent. Not only does the efficiency of any vessel vary from that of another; but, if you make a progressive trial, the curve of efficiency is not a straight line, but is varying also, which of course increases Mr. d'Eyncourt's difficulties. I do not quite know why Mr. d'Eyncourt has chosen his limiting economical speed at the fourth power. Perhaps I am not understanding the paper correctly, but I do not quite follow that. I might say, with regard to the Table on page 248, that I think he has not been generous enough to the full vessel. The little experience I have had with the experimental tank seems to point to the fact that the percentage should be a little higher. I do not know that I can add anything more, except to repeat what Mr. Hamilton has said, that it augurs well for the Institution that we find one of our younger members coming forward with such a paper as this.

Professor J. H. Biles (Member of Council): My Lord and Gentlemen, I am in much the same position as Mr. Denny. I should very much like to say something about this paper; I am very pleased we have a young member coming forward and bravely launching out into a difficult subject, full of intricacies, and willing to submit his results to the criticism of this Institution. I think his Table on page 248 would perhaps have been a little more rapidly understood, if he had assumed that we know a little less than he thinks we do. I have been trying to trace the connection between the length of the parallel middle body and the limiting speed, and I have had to go back to the old theory of Scott Russell to get the association properly in my mind in the short time at my disposal. The co-efficient of fineness as a prismatic co-efficient, represented in the left-hand column, is for the value 5. Practically, that value is satisfied if the curve of cross sectional areas consists of two triangles. I am sorry there is no blackboard here, or else I might have entertained you with a diagram; but the length of middle body increases the fulness in the proportion which is indicated, not by the figures 0, 14, 30, 49, but by 0, 10, 20, and 30, and the relation that Mr. d'Eyncourt points out there, as the result of his observation, would be exact, if, instead of being 86 it were 90, and instead of 68 it were 70, and instead of 48 it were 50. That is as I make it by the rough method and in the short time at my disposal. I think it is a curious and interesting result, that the limit of speed, which he has deduced from the results of trial trips, is so near to that which would be predicted by the old theory of Scott Russell. I am not sure whether I am correct, but I think that is so, and we should be much obliged to Mr. d'Eyncourt, personally, for having brought this subject forward in the manner that he has. I he could only see his way, when the paper is printed in the Transactions, to make it so that we could see the intermediate steps between the speed curves and the Table that he has given here, I think it would add considerably to the value of his paper.

Mr. E. H. T. d'Eyncourt (Member): My Lord and Gentlemen, first of all, I must thank the members for the very kind way in which they have received my paper. As time is short, and the subject rather a complicated one, I will ask to be allowed to send a reply to be published in the Transactions. I am sorry that Mr. Hamilton and Mr. Denny did not receive copies of my paper in time to look over it.

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The following communication, completing Mr. Tennyson-d'Eyncourt's reply, was received later:—

Referring to Dr. Elgar's remarks, there is, I think, no doubt that an experimental tank which would be at the disposal of all the shipbuilders in the kingdom, would be of the greatest value and assistance in determining the necessary horse-power, and suitable speed for proposed ships. That such a tank would be very largely used is certain; the only fear being that one tank might not be sufficient to accommodate all who might wish to have their models tried. Mr. A. Denny, in referring to my paper, said he was not quite clear as to why I had fixed the limiting economical speed as that at which the indicated horse-power is increasing as the fourth power of the speed. As I endeavoured to show in the paper, I took this point because it will be found possible to drive a longer vessel of greater displacement with the same horse-power as a shorter one, as soon as the speed is increased beyond this limit; and also, because the horse-power begins to go up in a very rapidly increasing ratio when the speed is increased beyond this limit. Mr. Denny also remarked that with their vessels, models of which had been tried in the tank at Dumbarton, the propulsive co-efficients were found to vary over a range of from 35 to 65 per cent. This, I think, must be taken to include very varying types of vessels, and the lower values of the propulsive co-efficient must be for low speeds, when the machinery is not working with efficiency; whereas the cases I have taken apply rather to the higher speeds, for which the machinery has been designed to give efficient results. Professor Biles in his remarks suggested that I should explain more fully the way in which I have arrived at the curves I have given. They have simply been deduced from actual results of progressive trials; taking the curves of indicated horse-power and speed, comparing them directly, and reducing them to a common standard, as shown in the paper.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Ladies and Gentlemen, I rise to propose a vote of thanks to the readers of those interesting papers which we have had to-day. You have heard already the very favourable criticisms which have been made upon Mr. d'Eyncourt's paper, criticisms which, I am afraid, I am not able to criticise; but I do feel that I am able to form an opinion upon those papers which have been so ably put before us by Lord Brassey, and I humbly think that he has done a great service to the country in bringing them forward at the time and in the way he has done. It is therefore my great pleasure to ask you to join me in a cordial vote of thanks to the readers of the papers on this occasion.

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## PRELIMINARY PROCEEDINGS,

### WEDNESDAY, JUNE 26, 1901.

THE following gentlemen, having been duly recommended by the Council, were unanimously elected Members of this Institution:—Mr. George Bartram, Partner in the Firm of Messrs. Bartram & Sons, Sunderland; Mr. Charles Brown, Junior Partner in the Firm of the Dock Ironworks, Bombay; Mr. George Bull, H.M. Dockyard, Chatham; Mr. James Cocks, Superintendent Engineer to Messrs. Huddart, Parker & Co., Ltd., Melbourne, Australia; Mr. Arthur Farquhar, Shipyard Manager to Messrs. Alexander Hall & Co., Aberdeen; Mr. A. Grant Gordon, Marine Superintendent and Consulting Engineer to the China and Manila Steamship Company, Ltd., Hong Kong; Mr. James Grier, Works and Drawing Office Manager to Messrs. John Readhead & Sons, South Shields; Mr. George Claude Hamilton, Departmental Manager to Messrs. Taylor Bros., Leeds; Mr. Edward S. Johnson, Barry, Cardiff; Mr. John Macwilliam, Surveyor to Lloyd's Register, Dublin; Mr. F. Wolff May, Director in the Firm of Messrs. Harland & Wolff, Belfast; Mr. David McGee, Manager to Messrs. John Brown & Co., Ltd., Clydebank; Mr. E. Richard Mumford, in Charge of Messrs. William Denny & Bros.' Experimental Tank, Dumbarton; Mr. George Nicol, Chief Draughtsman, the Northumberland Shipbuilding Company, Newcastle-on-Tyne; Mr. James Patterson, Director and General Manager to Messrs. Caldwell & Co., Ltd., Glasgow; Mr. Edward L. Peacock, of the Designing and Marine Engine Department of the Midvale Steel Company, Nicetown, Philadelphia, Pa., U.S.A.; Mr. J. Thomas Relf, Superintendent Engineer to the New Zealand Shipping Company, Leadenhall-street, E.C.; Mr. William Ross, Superintendent Engineer to Messrs. W. Thomson & Son, Shipowners, Dundee; Mr. James Russell, Manager to the Greenock Foundry Company, Greenock; Mr. Walter A. Sage, Marine Superintendent, Peel-street, Hull; Mr. Alexander W. Sampson, Shipyard Manager in the Shipbuilding and Engineering Company, Fairfield, Govan; Mr. Leslie Skinner, Manager and Director to Messrs. Wood, Skinner & Co., Ltd., Bill Quay-on-Tyne; Mr. Arthur D. Wedgwood, Partner in the Dennystown Forge Company, Dumbarton.

The following gentlemen were elected Associate Members:—Mr. Angus Brown, Junior, Partner in the Dock Ironworks, Bombay; Mr. John Robertson Dunn, Inspector of Machinery to Messrs. James Howden & Co., Glasgow; Mr. James Fernie, Consulting Engineer and Naval Architect, Liverpool; Mr. James Paul, Superintendent to the River Steam Navigation Company, Ltd., Calcutta; Mons. N. Polakov, Manager for Outside Marine Work to the Société des Chantiers Navals Ateliers et Fonderies at Nicholaieff, Russia; Mr. John R. Richmond, Managing Director to Messrs. G. & J. Weir, Glasgow; Mr. A. J. C. Robertson, of the Drawing Office of Mr. W. V. V. Ledgerwood, Marine Engineer, Bridge of Allan; Mr. David Showers, in Drawing Office of Messrs. John I. Thornycroft, Chiswick; Mr. F. Vernon Spence, Assistant Chief Draughtsman to Messrs. Sir W. G. Armstrong, Whitworth & Co., Ltd., Newcastle-on-Tyne; Mr. Alexander T. G. Thomson, of the Fairfield Shipbuilding and Engineering Company, Ltd., Govan, Glasgow; and Mr. William Weir, Managing Director to Messrs. G. & J. Weir, Ltd., Cathcart.

The following gentlemen were elected Associates:—Mr. I. J. Abdela, Mr. Joseph Beardmore, Lieutenant L. Beard du Dézert, Mr. J. C. Dobbie, Mons. Stephan Drzewiecki, Mons. J. Kannegiesser, Mr. J. W. Percival, Mr. William Reid, and Mr. J. H. Richardson.

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# ON FREEBOARD.

By Archibald Denny, Esq., Member of Council.

[Read at the Summer Meetings of the Forty-second Session of the Institution of Naval Architects, June 26, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.

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This short paper on freeboard is intended to be mainly historical, and not in any way controversial in its character. It is written because I felt that some of the work which has been done in connection with the existing Load Line Regulations should be preserved as being interesting in itself, and as placing on record data which may form useful points of departure in the future.

A great deal of valuable work relating to freeboard has been carried out by the Technical Staff of the British Corporation (with which society I have the honour to be connected), in connection with the Load Line Committees which have sat from time to time in recent years, and independent investigation has been made into the general laws governing the question. What is now placed before you is largely based upon that work.

The present Load Line tables are undoubtedly based upon Lloyd's Tables of Freeboard, first published in 1882, in which that Registry laid down the principle, subsequently adopted by the 1884-5 Load Line Committee, of reserve buoyancy as a basis of freeboard, and not height of platform *simpliciter*, as was strongly advocated by the Board of Trade. To Lloyd's Register, therefore, belongs the credit of evolving an orderly method of assessing freeboard out of the chaos which had previously existed. Without discussing in any way the correctness or the reverse of reserve buoyancy as a basis for freeboard, I submit a series of curves showing the method of its application.

Diagram No. 1 (Plate LI.) presents in a concrete form the different scales of freeboard under the Load Line Act for different classes of sea-going vessels, together with the Table A Freeboards of Lloyd's 1882 Tables, expressed in terms of length of vessel instead of depth, as tabulated in the Rules. The freeboard curves shown are all for vessels which have a depth equal to one-twelfth of their length; they are measured from the top of deck, and, taken in conjunction with the diagonal line, which shows the corresponding depth of vessels at any length, indicate the amount of free



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side in relation to depth required under the original Load Line Regulations of 1890, as well as under the different extensions, upwards and downwards, which were made subsequently.

Tables A and D give the respective freeboards for flush-deck steamers and sailing ships of full scantlings, and are based upon reserve buoyancy. These Tables provide the minimum height of platform permitted under the Load Line Act; no addition to structural strength justifies a reduction in freeboard in the case of a full scantling vessel; but, without adding to the scantlings of the main hull, deeper immersion is permitted on account of deck erections when fitted, provided these erections are of a certain defined character.

Table C, or the Table of Freeboards for awning-deck vessels, depends upon structural strength, and forms a convenient point of departure for the assessment of the value of erections which cover less than the whole length of the vessel.

Table B, for spar-decked vessels, is also based upon strength of structure, and was no doubt intended to be intermediate between the freeboards required for awning-deck vessels and the minimum freeboards which would be assigned to vessels having full scantlings up to the height of the superstructure deck, the curve for which is indicated by the dotted line at the top of the diagram. Table B, from several causes, is now practically inapplicable in its simplest form.

Diagram No. 2 (Plate LI.) embodies the result of investigations into the relation between the freeboard and reserve buoyancy figures given in the Tables as published in 1890. A mean line was run through the tabulated spots for reserve buoyancy for Table A (Steamers), and a model was found which agreed with the Table freeboard and the reserve buoyancy respectively, at 300 ft. length. Adopting the same form of model throughout, it was found that the agreement, not only with the tabulated freeboards but with the extensions of those freeboards to vessels having a depth one-tenth and one-fourteenth of their length respectively, was very close. This will be seen by reference to the diagram, where the crosses indicate the tabulated freeboards, and the curves show the actual freeboards which correspond with the mean curve of reserve buoyancy. The agreement is as close as could be expected without introducing an over-refinement of the length corrections given in the Tables.

Diagram No. 3 (Plate LI.) shows a similar analysis of Lloyd's 1882 Table A to that of the preceding diagram, with the difference that the process is reversed. Fair lines were run through the tabulated freeboards, and the true reserve buoyancies calculated for them, with the result shown in the diagram, that the reserve buoyancies to correspond with the tabulated freeboards are found to run in a straight line, and to give to all vessels of the same length the same reserve buoyancy. Evidently some slip

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had been made in framing this Table, as the fair curves of freeboard in the Tables could not be produced by the irregular line of reserve buoyancies tabulated. The reserve buoyancy figures used by Lloyd's, and the 1890 Load Line Tables are the same up to 270 ft. length, although the corresponding freeboards differ to some extent, as will be seen on referring to Diagram 1.

Diagram No. 4 (Plate LI.) shows fair curves of freeboard through the tabulated figures for Table D for vessels having depths respectively one-tenth and one-twelfth of their length. The calculated reserve buoyancies to correspond with these freeboard curves show the same result as for Lloyd's Table A; they lie along a straight line, and give all vessels of the same length the same reserve buoyancy. It is of interest to note that the freeboard curves last dealt with (Lloyd's Table A and Table D) hold a constant ratio to one another throughout their length. No analysis of Lloyd's 1882 Table D is necessary, as it is practically identical with the present Table D, except that a model of slightly different form has been used in obtaining the freeboards to correspond with the reserve buoyancy figures.

I think these diagrams prove clearly that the original framers of the Load Line Tables intended that all vessels of the same length should have the same reserve buoyancy, no matter what their depth might be, and this was their governing principle in their application of the Freeboard Tables to vessels of different proportions, the practical result being that the table length corrections give to a short vessel of a certain depth a much smaller reserve buoyancy than they do to a long vessel of the same depth.

When the Load Line Tables were first published, they ended with vessels of 34 ft. moulded depth, although there were even then in existence many vessels of greater depth, but as the number of these vessels increased from year to year, it was necessary to adopt some working method of dealing with their freeboard.

In 1892 the different assigning bodies came to a tacit agreement to extend the freeboard for large vessels in a straight line from the finishing point of the printed Tables: the effect of this is shown on Diagram 5 (Plate LII.). The greatest length correction given in the Tables was applied to all large vessels, although in the printed Tables this correction is a steadily increasing one up to 34 ft. depth: the effect was to make the extension of freeboards for large vessels of different proportions, a series of radiating straight lines, and to depart from the above stated principle of constant reserve buoyancy in all vessels of the same length, in that it gave to such vessels reserve buoyancies which varied with depth.

As it became obvious that the rate of increase both in freeboard and reserve buoyancy was too great for these larger vessels, the Load Line Commission of 1898, which was called primarily to adjudicate upon the question of the North Atlantic winter freeboard, was also asked to take up the question of the extension of freeboards to

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large vessels. After long deliberation it was decided to adopt for vessels of 38 ft. depth and upwards, having a length twelve times their depth, a uniform reserve buoyancy of 35·8 per cent., and a uniform correction for vessels of 34 ft. depth and upwards of 1·7 in. per 10 ft. of length. The freeboards and corresponding reserve buoyancies thus decided upon are shown on Diagram 6 (Plate LII.), where it will be seen that the freeboards converge and the reserve buoyancies diverge from each other, for the different proportions of depth shown, as the length of vessel increases. It will be observed that the middle curve for one-twelfth depth to length does not agree exactly with the spots for 35·8 per cent., but the difference was considered by the Committee to be so small as to be negligible.

I hope the members will agree with me that the labour expended on these investigations has not been wasted, and that it is desirable to place on record these facts in connection with the assignment of freeboard under Act of Parliament, which otherwise might be covered up by "use and wont." I have not dealt at all with the details of the Rules, such as allowances for sheer, round of beam, &c., but I may say that the allowances for departures from the normal are so arranged as to keep reserve buoyancy practically intact. Nor have I dealt with the modifications of the original Tables resulting from endeavours to legislate for special types, such as well deckers, raised fore deckers, sunk main deckers, &c. I desire, as I have said, to avoid anything controversial.

I have limited myself to laying before you certain facts relating to the groundwork of the Tables, which I trust may be found of some interest and value.

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## DISCUSSION.

Mr. Benjamin Martell (Vice-President): My Lord, Ladies, and Gentlemen, I have listened to Mr. Denny, I assure you, with a great deal of pleasure for many reasons. In the first place, it is such a new experience to see my friend Mr. Denny reading a paper of a non-controversial character, which does not admit of anything being said that will in the slightest degree disturb his equanimity; and, secondly, I am very pleased indeed to think that the lamp which was lighted by his esteemed brother, for whom we all had such an affection, Mr. Wm. Denny, who did so much in framing these tables, and who took such a great interest in them, should now be kept alive by Mr. Archibald Denny, who has given us the value of the experience he has gained since his brother's death. There is very little to be said on this paper, seeing that Mr. Denny has described historically the formation of these Rules, and it is done in such a manner that all who run may read. There is not much for me to say, although I may draw attention to the enormous amount of work, which Mr. Denny has alluded to, in framing these Load Line Tables. The Committee sat

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for eighteen consecutive months, and there was a large amount of discussion, as you can easily understand, in framing these Tables and in coming to a unanimous conclusion. It was said at the time, and chiefly by shipowners, although they now approve so highly of these Tables, "that it was perfectly impossible to make rules which should be applicable to all descriptions and sizes, and types of ships—that one could not take into consideration the proportions of length and depth and breadth —the round of beam and the size of the erections which tend so much to the safety of the ship, and that every ship should be dealt with on its merits." Well, there was a great deal to be said for that, particularly by those gentlemen who took a great interest themselves in the subject, and who loaded these ships very properly from their own practical knowledge. But there were a great many who, unfortunately, did not do this at that time, and a great many difficulties arose and actions were brought by the Board of Trade, and the shipowners were placed in a very awkward position in defending those actions. That led to this question being taken very seriously in hand, and I must say that, without wishing to be egotistical in the matter, but merely as one of the members of that Load Line Committee, I cannot help thinking that a great deal of credit was due to us, after sitting all that length of time, for framing tables which should ultimately be applicable to all descriptions and sizes of vessels—tables which, in the present day, have been so generally accepted that we now scarcely hear of any dissentients among shipowners, or any objections on their part to adopting them. I need not say that, in the first place, it has led to a great saving of life. The losses that used to occur from unknown causes, often attributed to the overloading of ships, have been reduced enormously; and if only we take that into consideration, it is a matter of the greatest credit to this country, to think that we were the first to undertake this great task, and that we have accomplished it in the way we have. Even the Germans and the Norwegians accept these Tables as a basis for the loading of their ships, and there is no doubt, from the correspondence I have had, that they will ultimately become general throughout the world. I have very little more to say on the subject, than to express my thanks to Mr. Denny for taking the trouble to bring before us the results of the labours of the Load Line Committee, and for placing on record the historical facts, to which we can refer at any time for the manner in which this great work was accomplished.

Dr. Otto Reiss (Visitor): My Lord and Gentlemen, I only got the paper a few moments ago, so that I have been unable to look through it as thoroughly as its most valuable contents deserves; but still, I should like to put one question to Mr. Denny. You are aware, as Mr. Martell has just pointed out, that the different Governments on the Continent, and the Societies interesting themselves in shipping matters, have had the question of fixing the freeboard under their consideration for these last few years. Of course there are some people who object very strongly to its introduction, while some people consider it very useful, just as it used to be in England, and as it still is now to a certain extent, I am told. Suppose now that the Government of any other country but England should pass, or intend to pass, regulations fixing the freeboard, there arises first this question:— The regulations must be made universal, in order that foreign vessels should not be stopped in a British port, because the new regulations of the foreign country will not be as effective as the British ones. In respect to foreign freeboard regulations, you have a paragraph in your law, in substance as follows: If these foreign regulations do not prove as effective as the British ones, as, for instance, if the freeboard mark is fixed too high in comparison to the British regulations, then the British Government officials will be entitled to stop the vessel when touching at a British port. In order to avoid such an unpleasant state of things, these regulations must have the same basis

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as the British ones; there is no other way. We on the Continent or anywhere else, must do that, because, if you take a lot of different bases, you will have different results. Now, I am told that some people in England think that the basis adopted is not the best one: I do not believe that. For myself, I think it is the best one to be had at present, but I should like to hear from a gentleman so well versed in the matter as Mr. Denny, whether you will stick to this basis for the future.

Mr. John Corry (Associate Member of Council): My Lord, Ladies, and Gentlemen, I cannot add much to the paper that has been read by Mr. Denny, but as a member of Lloyd's Committee who had the first inauguration of this very difficult question before them, and as one who, I assure you, devoted a long time to its consideration, I have at all times taken a great deal of interest in it. At first, it appeared to us practically an insoluble problem to fix a load line for all types of ships, but by the perseverance of our indefatigable chief surveyor, Mr. Martell, and his staff, and with the experience of the best shipowners in the country, the original Tables of Lloyd's load-line were produced. But, gentlemen, I may say that these Tables were not evolved from any purely mathematical or scientific formulæ; they were evolved from the experience of the ablest shipowners of the country; and they became the basis of the present Load Line Tables, which are practically, I believe, very slightly altered from the original Lloyd's Tables. I think it most important that similar principles should be adopted by all the Navies of the world, so that there should be no competition at the expense of safety, for we regard these Tables as giving a fairly safe load-line. They have been produced as the result of the vast experience, which I think all other nations must admit this country has had, with regard to the loading of ships in all climates, and in all parts of the world. The thanks of the whole community are mainly due to our friend Mr. Martell, for the indefatigable way in which he stuck to this most difficult and apparently impossible task, and I wish to record my appreciation of the value of the enormous work he did, more than to say anything else on this paper.

Professor J. H. Biles (Member of Council): I only wish to ask one question of Mr. Denny. I am sorry that someone else has not undertaken to do this, but the discussion has passed somewhat into the historical stage. These diagrams, involving as they do an enormous amount of labour, are deserving of the fullest consideration, but of course, it is impossible to give a great deal of attention to them in the time at our disposal. Diagram No. 6 (Plate LII.) gives some curves which appear to me to be very remarkable, and I should like to ask Mr. Denny if he is at liberty to say upon what basis the Committee arrived at the conclusion which is now shown, I think for the first time, in the diagram marked "Mean curve of reserve buoyancy." If you look at that curve, on Diagram No. 6, you will see that the percentage of reserve buoyancy increases gradually with the length of the ship, until, in the upper curve, we reach a length of 450 ft., and after that the percentage of reserve buoyancy falls off. I should like Mr. Denny, if he can, for the benefit of this audience, and in order to complete to some extent the history of this subject, kindly to explain on what basis the Committee went, in arriving at what appears to me to be such a remarkable conclusion. If you look at the lower curve, you will see that a 700 ft. ship has about the same percentage of buoyancy as a 520 ft. ship, and in the upper curve a 700 ft. ship has the same percentage of buoyancy as a 350 ft. ship. I do not know whether Mr. Denny has considered another question, and that is, whether the principle of percentage of surplus buoyancy is the right one to go upon in these very large ships. It seems to me that there must come a point where percentage of surplus buoyancy should not be the basis, but

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where, perhaps, a fixed minimum of absolute surplus buoyancy should be provided for a ship above a certain size, and that, beyond this, the increase of surplus buoyancy should not be necessary. There surely must come a size of ship where it is not necessary to increase the lifting power above water. I should be glad if Mr. Denny can give a reply to one, or both, of those questions.

Mr. Martell (Vice-President): Perhaps Mr. Denny might allow me to say a word or two in explanation. Professor Biles is quite right in drawing attention to this, but I may say that what guided the Committee originally—and I think Mr. Denny will bear me out in this, for although we could not act upon it fully at that time, it was altered afterwards—was this: that if you continued to increase the reserve buoyancy, if you got up to a ship of say 100 ft. in depth, you would have such an enormous amount of the hull of the ship out of water that it could not be utilised at all for the lifting power. Waves are only of a certain height, and if you get to the extreme height of waves as far as experience has shown, there is no amount of the hull of the ship out of water that could be utilised in any way, or do any good to the lifting power of the ship. As Mr. Denny said, there must be a point where you will have to draw the line, and that was the limit we fixed of 520 ft. for the length of ships. As he says, the depth of such a ship gives a certain amount of hull out of the water, which you can estimate by the reserve buoyancy, and any increase of depth beyond that can do no good whatever in increasing the lifting power. This is the basis of the principle of reserve buoyancy as laid down and framed in these Rules.

Mr. Archibald Denny (Member of Council): My Lord, Ladies, and Gentlemen, this paper, as Mr. Martell has said, was written so as to avoid controversy, and Dr. Otto Reiss has put me in rather an awkward predicament. I would crave the indulgence of the meeting if I do not express my personal views as to the rightness or wrongness of these Tables. I put them before you, and you can each draw your own conclusions. I would thank Mr. Martell also for the very kind reference he made to my late brother. His life and work have always been more or less of an inspiration to me, and I have endeavoured as far as possible to follow him. As to other Governments adopting our Load Line Regulations, that is really a matter for the other Governments and for the people of other countries to decide upon. We cannot easily get our Government to move—it never does want to move if it possibly can avoid it—and from my experience of Load Line Committees, I can quite understand the reason why they do not. Suppose now you said we want a new set of Tables, look at the vested interests you have to disturb; look at all the owners of vessels. You cannot make any possible alteration without affecting somebody; you will either increase or decrease somebody's freeboard. Even supposing you said: we will make new rules, but let all the older vessels keep their present freeboard until they sink or go aground, or are broken up; the new types, as soon as you introduced your new regulations, would sail as close to the wind as possible, and would probably attain greatly increased earning power, and to that the owners of the older boats would at once object. Therefore, gentlemen, you can quite understand that our Board of Trade has not the slightest desire to open up the discussion again. At the same time, there are the curves; if you look at them, you may perhaps be somewhat surprised that they should curve first one way and then the other; but do not forget, as I said in my paper, that, until Lloyd's came forward with their geometrical rule, there was perfect chaos, and even if we admit for a moment that reserve buoyancy is not the best basis, still the Rules seem to have given fair satisfaction. Some of us have the idea that perhaps the flush-deck vessel without any erections does look, when you see her loaded and going down the river Clyde, a little like a sand barge, but then it is not the practice to

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have vessels with no erections nowadays, and the vessel with erections, I think you will all admit, looks very well. Professor Biles asked a question as to the divergence of the curves. In committees, as he no doubt very well knows, a certain amount of compromise has always to be made, and the difficulty we had was that the terms of submission from the President of the Board of Trade did not admit of our revising the old rules—we were to extend, and not to revise. After we showed that you could not possibly join a straight line on to a curved one without some corner in it, the Board of Trade did allow us to modify to a very slight extent, but it was only to a slight extent. The idea was just as Mr. Martell has said, that after reaching a certain size, vessels practically neither require additional reserve buoyancy, nor, perhaps, even additional freeboard. It is quite evident that if you could build a steamer so that she had sufficient stability, and you absolutely prevented the water from getting down below, she could have a very small freeboard indeed, but then you have the comfort of the passengers and crew to look to, and that is why the Board of Trade were so insistent upon height of platform as a great element. Practically, what did result from the rules of Mr. Martell, and Mr. Reed, and Mr. Jenkins—I may perhaps be allowed to mention their names—was that they gave a height of platform simply in another form, but one in which they could, by proper calculation, pass from one vessel to another, with some sort of reasonable rule to guide them. I do not know that I can answer Professor Biles's question further, except to say that if you stick to a constant reserve buoyancy of 35·8 for $\frac{1}{12}$th depth, the constant length condition is bound to give you diverging lines of reserve buoyancy for $\frac{1}{10}$th and $\frac{1}{14}$th depth to length. The idea the Committee had was that they could not say, without further experience of these large vessels, what would be the smallest limit of freeboard, but they could safely say, when the length became very great, that the probability was that the length was the governing factor, and that the freeboard should be the same irrespective of length.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Ladies and Gentlemen, I think you will all agree that we have had a most interesting paper from Mr. Denny. It touched, perhaps, on rather delicate matters, but he has dealt with them in an exceedingly able and pleasant manner, and I am sure you will all join with me in moving him a vote of thanks for his paper. I will now ask Mr. Yarrow to be good enough to make a statement which he wishes to be bring forward.

### Mr. Yarrow's Proposal for an Experimental Tank.

Mr. A. F. Yarrow (Vice-President): In the course of the remarks made by Dr. Elgar yesterday, it will be remembered that he referred to a tank which had been established at Washington by the Government, for testing the resistance of models. He also told us that the Navy Department at Washington not only made use of this tank for their own investigations, but that it was thrown open to the shipbuilders of the United States for the purpose of making tests, under the direction of the persons in charge; so that, in this way, whatever benefits might accrue from tank experiments are obtainable by the shipbuilders in the United States. As you all know, the British Admiralty have a tank of their own, and I have always understood that it has been found of very great service. Messrs. Denny have their own private tank, and Mr. Denny can inform us whether they have found that the researches they have been able to make with it have been of practical utility. We also know that numerous tanks have been provided by various Admiralties in different parts of the world. Now, I would submit that, in view of the rapid increase in competition in our profession, it behoves us to advance with the times; no stone should therefore be left unturned, by shipbuilders in this

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country, to keep well up to the mark, and all the facilities which modern appliances can give should, if possible, be available to them. I would therefore submit to the consideration of the Institution of Naval Architects whether there could not be established under its auspices a tank which might be available, not only for shipbuilders of this country and for members of this Institution, but for anyone willing to pay for the information obtained. It might be possible to establish such a tank in the Midland Counties, so that it would be equally accessible to all shipbuilding centres. It should, I presume, be under the charge of a thoroughly competent person, who would be bound to absolute secrecy as to the results obtained for private firms. I am not so sanguine as to expect—at any rate in the early stages—that the charges made for investigations will pay a remunerative interest on the outlay; but it is not unreasonable to expect that some return might be obtained. It must be borne in mind that one of the objects of the Institution is to assist research in the science of naval architecture; such a proposal, therefore, comes quite within the scope of the Institution. Having in view the educational advantages for naval architecture that are obtainable in the United States and in Germany, such a tank might be made available (if independent of any particular University) for all educational establishments where naval architecture may, in the future, be taught. I think it would be desirable if some expression of opinion could be obtained from the present meeting on this subject, and if it were deemed desirable to carry out such a scheme as now suggested, I would propose a motion to the effect :—" That this meeting, having regard to the desirability of establishing a tank in this country for testing the resistance of models, which might be available for all shipbuilders, requests the Council of the Institution to take the matter into serious consideration with the view of arriving at the best means of carrying out the suggestion."

Mr. Benjamin Martell (Vice-President) : My Lord, Ladies, and Gentlemen, I think the suggestion of my friend Mr. Yarrow is well worthy of consideration. I do not think there would be any difficulty in providing the funds for the erection of such a tank, because I may adduce an instance where a wealthy gentleman in Glasgow—in this very city—consulted me with reference to providing such a tank himself, and I gave him, I need not say, my very hearty support, and trusted that he would carry his scheme out. However, for some reason or other—I could never ascertain why—that fell to the ground, and he did not do it. Now, I think such a suggestion as Mr. Yarrow has made, that a tank should be erected, like the one in America, where it could be placed under an independent board, and with an entirely independent man to preserve the results of experiments, and hand them to those who employed him, would meet with general approval; and I do not think there would be great difficulty in finding gentlemen who would contribute towards the establishment of such a tank. I have no doubt that in the course of a short time it would be made remunerative, not only so as to pay its own expenses, but also to return something to the Institution of Naval Architects for the trouble they may take in superintending it, and appointing a person in the same way as Lloyd's Register did when they were entrusted by the Government with the task of superintending the testing houses for cables and anchors. This has been conducted in the most satisfactory manner, and those testing houses have been made to pay, and at the same time Lloyd's have had sufficient to remunerate them for the trouble that they have taken in superintending them. Therefore, my Lord and Gentlemen, I give this suggestion my most hearty approval, and trust that it will be brought to a successful issue.

Sir Nathaniel Barnaby, K.C.B. (Vice-President): My Lord, Ladies, and Gentlemen, all those present who are in any way familiar with shipbuilding and the designing of ships, know how very much our profession is indebted to the late Mr. Froude. There is no one who, during the last one hundred



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years, has contributed so much to the advancement of our profession as did Mr. Froude, and his great work was to change our ways of looking at the laws of fluid resistance. The establishment of his tank was due to the fact that Sir Edward Reed was then Chief Constructor of the Navy, and he was so deeply interested in what Mr. Froude showed him of what he had done by his own private means, that Sir Edward recommended strongly to the Admiralty that the tank at Chelston Cross, Torquay, which Mr. Froude had established, should have Government support. For some years it was always a difficulty to get the Government subvention for that, but now it is a firmly established fact, and no new design ever passes out of the Admiralty that has not been subjected to trials in this tank. The other day I was anxious to know Mr. Denny's feeling concerning the usefulness of the tank in his private works. He was good enough to send me a copy of his paper, which I had not seen, in which he speaks in the very highest terms of the value which these investigations have been to his firm. When Mr. Yarrow rose this morning, I had no idea as to what he was going to propose, but I felt, when I heard him, that it was the right thing. We shall make this meeting truly famous: yesterday we did a great thing, and if we can do what Mr. Yarrow proposes to-day, we shall accomplish another great object, which I entirely support.

The Right Hon. Lord Brassey, K.C.B., D.C.L. (Past President): My Lord, Ladies, and Gentlemen, I entirely follow the scientific men who have addressed us, with regard to the value and, indeed, the necessity of a tank such as has been referred to in Mr. Yarrow's address. The only question that I ask myself is this: Is it not possible that the tank which is under the administration of the Admiralty should be made available for the investigations of private persons? If it is not fully employed, if the staff have available time for such work, I fail to see any difficulty in making some arrangement by which the existing tank should be made more generally useful. If, however, the Government resources are fully employed in Government work, the question lapses, and there is nothing more to be said. We then require to carry out the suggestion of Mr. Yarrow in its integrity. I do not know—perhaps Sir Nathaniel Barnaby can tell us—whether the Government staff are fully employed on their own work, because, if they are, then it is necessary to do something else; but, if they are not fully employed, we ought to look to them to co-operate in a great national work.

Sir Nathaniel Barnaby: May I say, speaking as an old Admiralty official, that, although they might possibly undertake more work, it is a very undesirable thing to distract the attention of the gentlemen who are working there constantly in making investigations, not only on actual models, but concerning principles. Mr. Froude, the son of the late Dr. Froude, is in charge there, and it is better not to disturb him in his work; but, if possible, to do what I believe we can do perfectly well. If the Council of this Institution will only put its shoulders to the wheel, we shall make it go through.

Mr. Francis Elgar, LL.D., F.R.S.: After the reference I made to this subject yesterday, in speaking on Mr. d'Eyncourt's paper, it is hardly necessary for me to say now that I very strongly support the proposal of my friend Mr. Yarrow, and I would ask the meeting to give his motion its very hearty support. I am quite sure that if the Council feel they have the general support of the members of the Institution in considering such a step as that proposed by Mr. Yarrow, it will be possible for them, in due time, to put some practical proposal before the Institution. It might perhaps be done partly in association with the Admiralty. We do not know that it could not. I quite agree with Sir Nathaniel Barnaby as to the desirability of not interfering at all with the Admiralty, if it could possibly be avoided; but if we were to have an interview with the Admiralty authorities, and to ascertain what their position is, and what their views are with regard to the matter, I do not know

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that we might not get some good out of it. Perhaps we could succeed in getting them to see the desirability of sometimes giving help to private shipbuilders in exceptional cases, provided they were not likely to be frequently troubled. I will ask the meeting to give a very hearty support to Mr. Yarrow's proposal, in order that the Council may feel authorised to take this question seriously in hand.

Lord BRASSEY: Having heard the explanation given to us by Sir Nathaniel Barnaby, I desire to say that I entirely support Mr. Yarrow's proposal.

Mr. THOMAS: A remark from one coming from the United States will not, perhaps, be out of place. As to the tank at Washington, it is open to the shipbuilders of the country at large. There is a sort of introduction required, and a small charge is made in order to defray expenses, and to put the thing on a proper footing, but the policy of Admiral Melville and his staff has been that the tank and the Navy Department have been paid for by the support of the nation at large. Whatever scientific results have been obtained from its investigations are rightfully the property of the people, who should have, under proper control, access to these results. I have in my mind one case in which a gentleman of considerable means in the United States came to a prominent firm of shipbuilders and proposed a certain type of vessel. He was a man of great intelligence, and had spent much time in investigating the problems concerning a certain form of ship which reverted to the "cod's head and mackerel's tail" type, which we all know about. The firm of shipbuilders did not consider it would be wise to put the proposed amount of money into the ship, and for that reason they suggested that he might make investigations at the tank at Washington. The curves that were plotted showing the resistance of this model, and the others which had been made formerly in the United States, convinced this gentlemen at once that it would be folly to put say, 500,000 dollars into the steamer he had proposed. I mention that as an instance of the value of this tank. I should be very happy to see the private concerns of Great Britain have such a tank at their disposal, so that they could test not only Government boats, but private boats.

The PRESIDENT: I am afraid, as time is so short, we can hardly go into any further discussion.

Mr. ARCHIBALD DENNY: My Lord, Ladies, and Gentlemen, perhaps I may be able to help the meeting by telling them that, had we two tanks, we could use them both, and it seems to me that one tank will scarcely satisfy all the shipbuilders of this country.

The PRESIDENT: I will now proceed to put Mr. Yarrow's motion to the meeting.

Mr. Yarrow's motion, having been put to the meeting, was carried unanimously.

The PRESIDENT: Ladies and Gentlemen, I have much pleasure in rising for the purpose of telling you that our German fellow members and colleagues, with great good feeling, desire, on the occasion of their presence in Scotland, to send a telegram expressing their gratification to our King. We naturally wish to associate ourselves with them in this, and also to join with them in sending to His Majesty the German Emperor a telegram expressing the satisfaction it affords us to work in friendly harmony with our German colleagues, and the pleasure it gives us to recall His Majesty's gracious reception of our Institution in Berlin. I will now read the telegrams which it is proposed to send:—

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*To* His Royal and Imperial Majesty Edward VII., King of Great Britain and Ireland,
*Marlborough House, London.*

The Institution of Naval Architects and their guests, the German Schiffbautechnische Gesellschaft, assembled in Summer Meeting at Glasgow, desire with humble duty to send their respectful greetings and warm wishes for happiness to your Majesty.

The members of both Institutions remember with gratitude your Majesty's Honorary Presidency, as Prince of Wales, of the International Congress of Naval Architects and Marine Engineers in 1897, and the gracious reception which her late lamented Majesty, Queen Victoria, gave to the Congress at Windsor Castle.

The members assure your Majesty that they are working in friendly harmony for the improvement of shipbuilding and the advancement of international commerce.

(Signed) Earl of Glasgow,
*President Institution of Naval Architects.*

Professor Busley,
*Schiffbautechnische Gesellschaft.*

*Corporation Galleries, Sauchiehall Street, Glasgow.*

*To* His Imperial and Royal Majesty William II., German Emperor, King of Prussia,
*Kiel, Germany.*

The Institution of Naval Architects and their guests, the Schiffbautechnische Gesellschaft, assembled at their Summer Meeting in Glasgow, desire with humble duty to send their respectful, greetings and warm wishes to your Majesty and, at the same time, to inform your Majesty that they are working in the most friendly and harmonious co-operation for the improvement of shipbuilding and the advancement of international commerce.

The members of the Institution of Naval Architects recall with utmost gratitude the kindness they received from your Majesty in 1896, and also your Majesty's memorable words: "Blood is thicker than water."

(Signed) Earl of Glasgow,
*President Institution of Naval Architects.*

Professor Busley,
*Schiffbautechnische Gesellschaft.*

*Corporation Galleries, Sauchiehall Street, Glasgow.*

I beg to move that these telegrams be sent this forenoon, and I hope that we shall get a reply in the evening.

The motion, having been put to the meeting, was carried unanimously.

During the dinner held at the Grosvenor Restaurant, the same evening, the following replies were received:—

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*Buckingham Palace*, 7.34 *p.m.*

*To* Lord Glasgow, *Grosvenor Restaurant, Exhibition Grounds, Glasgow.*

I am commanded by the King to thank the Institution of Naval Architects and their guests, the German Schiffbautechnische Gesellschaft, for their kind telegram and good wishes. It affords His Majesty much pleasure to hear that the two bodies are working together so harmoniously for the common good.

Knollys.

*Kiel*, 9.35 *p.m.*

*To* P. C. Earl of Glasgow, *President Institution of Naval Architects, Corporation Galleries, Glasgow.*

The telegram sent by you as President of the Institution of Naval Architects and by their guests, the Schiffbautechnische Gesellschaft, assembled at Glasgow, has given me much pleasure, and I beg you to express to them my warmest thanks for their kind wishes. The promotion of shipbuilding and of international commerce, fit to bring closer together kindred nations moved by the same spirit of enterprise, has, as you know, all my sympathy, and I wish every success to the useful and noble work in which you are engaged.

William I. R.

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# THE TRANSVERSE STRENGTH OF SHIPS.

By J. Bruhn, Esq., B.Sc., Member, Surveyor to Lloyd's Register of Shipping.

[Read at the Summer Meetings of the Forty-Second Session of the Institution of Naval Architects, June 26, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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## Frames, Floors, and Deck Beams.

In a paper read before this Institution in 1882 by the late Messrs. Read and Jenkins, attention was called to the fact that the question of the transverse strength of ships had not received the consideration it deserved. The remark is perhaps as true now as then. The longitudinal strength has come to be looked upon as the strength of a ship to such an extent that, whenever the calculated strength of a ship is mentioned, no other is usually thought of. The longitudinal bending moments being so much larger than the transverse ones, it follows that the longitudinal strength is more important than the transverse, but it is so only in regard to the quantity of material used in the construction. From the point of view of the safety of the ship and cargo, the strength of no one part of the structure can claim to be more important than that of other parts. If the cargo or ship is damaged, it matters little whether the cause is longitudinal or transverse weakness. It is, therefore, desirable to have a method whereby the transverse strength of a ship can be estimated as exactly as the longitudinal strength, or practically so. The object of the present paper is to attempt to provide at least another step towards the establishing of such a method.

The distribution of the material in the longitudinal girder is comparatively simple. The breadth, depth, and form of the girder being practically given, the strength can only be varied by modifying the thicknesses of the plating. It is otherwise with the transverse material. Here the efficiency, or the strength, lightness, and compactness of the structural arrangement, depends to a much greater extent on the form and distribution of the material. We have in this case much more freedom in the design of the structural parts. It is, therefore, all the more important to have a method whereby the stresses can be estimated; and as, moreover, the case is somewhat more complicated than that of estimating the longitudinal strength, the chances are that the employment of a scientific method will detect greater room for improvements in design in this respect.

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It may be desirable to briefly state what is understood in this paper by the expression athwartship strength.

The transverse stresses are those tending to change the form of the cross sections of the vessel. They are directly due to transverse forces acting on the ship, and, indirectly, to longitudinal forces.

The transverse forces are :—

(1) The weights of the structure and cargo.

(2) The reactions of these weights due to changes in the motion of the vessel (rolling, pitching, or heaving).

(3) The externally applied forces, such as the pressure of water, keel blocks, or other supports.

The longitudinal forces that affect the transverse strength of a ship are chiefly those which bend the vessel in a fore and aft plane, and, consequently, tend to deform the transverse sections.

Of the transverse forces the weights can be estimated exactly, and their reactions can also be found, when the conditions of rolling, pitching, or heaving are known. The external water pressure can be calculated, when the pressure per unit area is known. In still water it can, therefore, be estimated as accurately as the weights of the structure, but in agitated waters it can only be determined by experiments either direct and exact, if the method is scientific, or indirect and rough, if the results are found by the failures and successes in practice of actual structures. The rigid supports such as those due to keel blocks, cannot, however, be determined exactly, without knowing the conditions of the straining of the vessel.

Taking, then, a vessel floating freely in the water, we can determine all the forces which act on the structure, when the conditions are stated. The forces are of a simple nature. They are, in fact, in most cases reduced to a pure normal pressure, and the structure need only be designed to resist such forces. The difficulty is, however, usually to determine what will be fair conditions to assume for the worst possible case which the structure must be supposed to meet.

These conditions given and the forces estimated, there still remains the difficulty of apportioning the work of resisting these forces to the various parts of the structure. The longitudinal bending moments can be estimated by statical methods, when the conditions as to forces are given. In the case of the bending moments on the transverse parts of a ship, such as frames, beams, and floors, the case is different. The structure is from this point of view of a redundant nature, and the stresses consequently indeterminate by the usual statical methods. That is, the work of the

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various parts of the structure may be distributed in several ways, and yet satisfy all the ordinary conditions for equilibrium. It is, therefore, necessary to have one or more additional conditions, which must be fulfilled, before it is possible to determine the straining forces on the structure.

For convenience in the examination of the transverse strength of a ship, we may assume a section, say, one frame space in length, cut out of the vessel, and then we may apply all those forces to it which are necessary to keep it in equilibrium under the given conditions. Let Fig. 1 (Plate LIII.) represent such a cross section of a simple case, that of a small single-deck vessel, without pillars, floating freely in still water. This ring must be in equilibrium. The weight of the structure and the cargo, the pressure of the water, and the bending and direct stresses on the ring, must, therefore, balance each other. Let the weight of the ring and the corresponding cargo be equal to the displacement. The vertical main forces are thereby balanced. The pressure of the water on the two sides of the vessel will of course always be equal in still water. The ring, as a whole, is therefore in equilibrium. If we assume the deck beam removed, then the structure is simply firm, and the stresses are directly determinable by statical means. When, however, the beam is fitted, then there will be a force and a bending moment at G, which will resist part of the forces, but how large a part cannot be directly determined by the ordinary conditions of moments.

Messrs. Read and Jenkins in their above-mentioned paper adopted, as the additional requirement necessary for the solution of the problem, the conditions of continuity in the deflexions of the structure, and they determined by these means the forces in the case of vessels resting on keel blocks in dry dock.

The most direct method for the solution of a problem of this nature is, however, probably that based on the principle of least work. The systematic application of this principle to engineering problems is mainly due to Alberto Castigliano, of Turin. His methods have been extensively employed by civil engineers on the Continent; but have not, as far as I am aware, been applied to ship calculations, although exceedingly useful, not only in calculations for the strength of the main structure of a ship, but also in the determination of the strength of many details of the structure, such as rudders, stern forgings, masts, and rigging, &c. The very general character of the method, and the possibility of employing the ordinary rules for approximate integration, make it peculiarly applicable to the unsymmetrical structure of a ship.

Let any possible support which the section, shown in Fig. 1 (Plate LIII.), might get from adjoining sections be neglected. The ring is, therefore, only subjected to direct forces and bending moments in the plane of the sketch. At the point N there will thus be a direct force, say, F, a shearing force R, and a bending moment M, and there will be no other straining agencies on a cross section at this point. Let the horizontal or direct

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force at a section at K, the middle of the floors, be $P_0$, the shearing or vertical force $Q_0$, and the bending moment $M_0$. The forces and bending moments at N must be the same as those at K, with the addition of those acting on the intermediate part of the girder. Let $v$ represent the vertical pressure of the water on the part K N, $h$ the horizontal pressure, $c$ the vertical pressure, or weight of the cargo, and $s$ the weight of the structure. We have, then, if $a$ be the angle which a tangent to the section at N makes with the horizontal, and V, H, S, and C, the moments about N of $v$, $h$, $s$, and $c$ respectively, that—

$$F = P_0 \cos a + Q_0 \sin a + v \sin a - h \cos a - (s + c) \sin a. \quad (1)$$

$$R = - P_0 \sin a + Q_0 \cos a + v \cos a + h \sin a - (s + c) \cos a. \quad (2)$$

$$M = M_0 + P_0 y - Q_0 x - V - H + S + C. \quad (3)$$

It will be seen that the forces and the bending moment acting at N can be determined from the above equations, if the corresponding quantities are known for the point K.

It is a universal law, that whenever a structure is strained, the work done thereby is a minimum, or, in other words, the structure adapts itself with the minimum of effort to resist the applied forces. The work done in straining a structure is easily determined in simple cases. If $p$ is the stress per unit of sectional area, and E the modulus of elasticity, then the general expression for the work done will be—

$$W = \int\int\int \frac{p^2}{2E} d y \, d z \, d x.$$

For a bar subjected to direct pull or push stresses the work is therefore—

$$W = \frac{1}{2E} \int \frac{F^2}{A} d l,$$

where F is the total force, $l$ the length of the bar, A the sectional area, and E the direct modulus of elasticity. If the section is uniform, then—

$$W = \frac{F^2 l}{2 A E}.$$

For a bar subjected to shearing stresses—

$$W = \frac{I}{2G} \int \frac{\mu R^2}{A} d l,$$

or, if of uniform section—

$$W = \frac{\mu R^2 l}{2 A G}.$$

R being the total shearing force, $l$ the length of the bar, A its sectional area, G the shearing modulus of elasticity, and $\mu$ a co-efficient, which would be unity, except for the fact that the shearing tresses are not, like the direct stresses, uniformly distributed



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over the area of a cross section. The value of $\mu$ can always be determined for any section, when the distribution of shearing stresses is known. We have—

$$\mu = \frac{\iint q^2 \, dy \, dx}{q_1^2 \, A},$$

where $dy \times dx$ is an element of sectional area, $q$ the actual shearing stress per unit area, and $q_1$ the total shearing force on the section in question divided by the total sectional area A. $\mu$ is always larger than unity, but, usually, by so small an amount that it may, in most cases, be assumed equal to one without serious error.

For a bar subjected to bending stresses, we have—

$$W = \frac{1}{2E} \int \frac{M^2}{I} \, dl,$$

where M is the bending moment, I the moment of inertia of the cross section, $l$ the length of the bar, and E the direct modulus of elasticity. For a uniform bending moment, we have—

$$W = \frac{M^2 l}{2 I E}.$$

The total work done over the whole of the transverse girder shown in Fig. 1 by direct pull and push stresses, shearing and bending stresses, will be—

$$W = \frac{1}{2E} \int \frac{F^2}{A} \, dl + \frac{1}{2G} \int \frac{\mu R^2}{A} \, dl + \frac{1}{2E} \int \frac{M^2}{I} \, dl.$$

W must, by the principle of least work, be a minimum with regard to the applied forces. F, R, and M may, by equations (1), (2), and (3), be expressed in terms of the only unknown quantities $P_0$, $Q_0$, and $M_0$. The work done by the straining forces must, therefore, be a minimum with regard to $P_0$, $Q_0$, and $M_0$; or the rate of increase of work done with regard to these quantities must be zero. We have therefore :—

$$\frac{dW}{dP_0} = \frac{1}{E} \int \frac{F}{A} \frac{dF}{dP_0} \, dl + \frac{1}{G} \int \frac{\mu R}{A} \frac{dR}{dP_0} \, dl + \frac{1}{E} \int \frac{M}{I} \frac{dM}{dP_0} \, dl = 0 \qquad (4)$$

$$\frac{dW}{dQ_0} = \frac{1}{E} \int \frac{F}{A} \frac{dF}{dQ_0} \, dl + \frac{1}{G} \int \frac{\mu R}{A} \frac{dR}{dQ_0} \, dl + \frac{1}{E} \int \frac{M}{I} \frac{dM}{dQ_0} \, dl = 0 \qquad (5)$$

$$\frac{dW}{dM_0} = \frac{1}{E} \int \frac{F}{A} \frac{dF}{dM_0} \, dl + \frac{1}{G} \int \frac{\mu R}{A} \frac{dR}{dM_0} \, dl + \frac{1}{E} \int \frac{M}{I} \frac{dM}{dM_0} \, dl = 0 \qquad (6)$$

F, R, and M having the values given in equations (1), (2), and (3), we can determine $P_0$, $Q_0$, and $M_0$, from equations (4), (5), and (6), and then the forces and bending moments acting at any point of the girder may be found. The problem of determining the straining agencies by the principle of least work is thus solved in theory at least.

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A simple examination will show that whenever the span of a girder is considerably in excess of the depth, as will be the case in most engineering problems, the work done by the shearing forces is infinitesimal compared with that done by bending moments, and the second term in the expressions for $\frac{dW}{dP_0}$, $\frac{dW}{dQ_0}$, and $\frac{dW}{dM_0}$ may, therefore, be neglected without any error of practical importance.

Moreover, the first term is also of no practical importance in most problems in connection with the transverse strength of ships, although in many other problems it may be of the greatest importance. It might be expected that the work done by the direct forces would be small in the case of a girder without any rigid supports, such, for instance, as those forming the abutments of arches. The corresponding term in the estimate of the work done has, therefore, been neglected in the following calculations, but it can be included without much difficulty wherever it is necessary.

There thus remains only the last term to be considered, and, by equation (3), we get—

$$\int \frac{M}{I}\frac{dM}{dP_0}\,dl = \int \frac{1}{I}(M_0 + P_0 y - Q_0 x - V - H + S + C)\,y\,dl = 0 \qquad (7)$$

$$\int \frac{M}{I}\frac{dM}{dQ_0}\,dl = \int \frac{1}{I}(M_0 + P_0 y - Q_0 x - V - H + S + C)\,x\,dl = 0 \qquad (8)$$

$$\int \frac{M}{I}\frac{dM}{dM_0}\,dl = \int \frac{1}{I}(M_0 + P_0 y - Q_0 x - V - H + S + C)\,dl = 0 \qquad (9)$$

The practical integration of these terms through the length of the girder considered may be performed by any of the approximate methods of integration. Simpson's rule, being the one usually employed in shipyards, may be conveniently adopted here. The girder to be considered is the whole of the transverse ring; but, as it is symmetrical about the middle line, we need only consider the one half. The work done by the various forces on the part K G L (Fig. 1, Plate LIII.), must, therefore, be a minimum. At G there will be a point of discontinuity, and it is, therefore, desirable to make the integration over K G and G L separately. The girth of the section from K to G and from G to L is, therefore, divided into an even number of equal intervals, and the moments of inertia of the cross sections of the girder at these points are calculated. The values of $y$ and $x$, corresponding to the moments at $P_0$ and $Q_0$ respectively, are recorded, and the moments V, H, S, and C determined. The last four quantities being all known, might at once, if found convenient, be added up. We can then write down for each point on the girder an expression containing four terms, of which the three first contain the three unknown quantities $M_0$, $P_0$, and $Q_0$. We may deal with all the terms as if they were known, put them through Simpson's rule, and multiply by the interval. If the results found for K G and G L be added, and the whole equated to zero, then we shall have the condition required by equation (9) fulfilled. It is not

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necessary to multiply by $\frac{1}{3}$ the interval, as is usual, as the result is to be equated to zero, and if the intervals of K G and G L are the same, it is not necessary to perform any multiplication at all.

If now each term, when multiplied by Simpson's multipliers, be further multiplied by the corresponding values of $y$, the results be added up, multiplied by the intervals, as before, and equated to zero, then we have the second condition corresponding to equation (7).

Further, if the same terms be multiplied by $x$, instead of by $y$, added, multiplied by the intervals, and equated to zero, we shall get the third condition required by equation (8). The sign of $x$ is of no importance, as the result is to be equated to zero.

We thus have the three equations from which $M_0$, $P_0$, and $Q_0$ may be found, and, when they are determined, we can calculate the bending moment at any of the selected points on K G L.

It may be desirable to take a numerical example to more clearly show the method adopted. It will be observed that on the sketch shown in Fig. 1 (Plate LIII.) it was assumed for simplicity's sake that no pillars were fitted. In that case considerations of symmetry would show that there could really be no shearing force at K, and there would thus be only two unknown quantities to be determined, and only two equations are required. When a row of pillars is fitted at the middle line, as assumed in Fig. 2, then there may be a shearing force immediately at the side of them, and the problem is as already explained.

Fig. 2 (Plate LIII.) represents the midship section of a small sailing vessel of 20 ft. breadth, 11·58 ft. depth, and 9·6 ft. draught. For the present investigation, we may assume the section to be 1 ft. in length. The displacement of one side of the vessel is 5,170 lbs. per ft. length of ship, the weight of the structure 1,500 lbs., and the cargo carried 3,670 lbs. per ft. length. The girth from 0 to 8 is divided into eight equal intervals, 2·310 ft. in length, and the length of the half beam is divided into four intervals 2·475 ft. in length. The moments of inertia of the various cross sections of the girder are then calculated for a frame space, and divided by the length of the frame space, in this case 1·75 ft., in order to obtain the moment of inertia correspond ing to 1 ft. length of ship.

Table I. (page 227) is then formed with six columns, marked M, P $y$, Q $x$, S, C, and (H + V) respectively, and with the number of the points or sections considered marked to the left of the table. The column marked M gives the bending moment at section 0. Column P $y$ gives the moment of the horizontal force at 0 about the various

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points. Column Q $x$ gives the moment of the vertical force at 0. Column S gives the moment of that part of the structure which lies between 0 and the point considered. The next column gives the same figures for the cargo, and the last column gives the total moment of the horizontal and vertical pressures of the water.

The numbers in the last three columns represent foot pounds, and might, therefore,

TABLE I.

| No. of Section. | M. | P $y$. | Q $x$. | S. | C. | (H + V). |
|---|---|---|---|---|---|---|
| 0 | + 1 | + ·00 | − ·00 | + 0 | + 0 | − 0 |
| 1 | + 1 | + ·60 | − 2·22 | + 172 | + 1,080 | − 1,620 |
| 2 | + 1 | + 1·20 | − 4·44 | + 695 | + 4,200 | − 6,300 |
| 3 | + 1 | + 1·80 | − 6·66 | + 1,530 | + 9,150 | − 13,550 |
| 4 | + 1 | + 2·75 | − 8·75 | + 2,620 | + 15,400 | − 23,700 |
| 5 | + 1 | + 4·65 | − 9·95 | + 3,700 | + 19,400 | − 32,700 |
| 6 | + 1 | + 6·95 | − 10·00 | + 3,700 | + 19,400 | − 38,700 |
| 7 | + 1 | + 9·25 | − 10·00 | + 3,700 | + 19,400 | − 45,300 |
| 8 | + 1 | + 11·58 | − 9·90 | + 3,700 | + 19,000 | − 51,500 |
| | | | | | | |
| 8 | + 1 | + 11·58 | − 9·90 | + 3,700 | + 19,000 | − 51,500 |
| 9 | + 1 | + 11·75 | − 7·45 | + 1,035 | + 10,080 | − 39,300 |
| 10 | + 1 | + 11·85 | − 4·95 | − 1,910 | + 990 | − 26,790 |
| 11 | + 1 | + 11·98 | − 2·48 | − 5,100 | − 8,050 | − 14,300 |
| 12 | + 1 | + 12·00 | − ·00 | − 8,600 | − 17,200 | − 1,400 |

at once be contracted to one column, if it is only a question of determining the bending moments for the one set of conditions. If the moments are also desired for altered conditions, for instance, with regard to the distribution of cargo, or the amount of draught of water, then it is convenient to keep these columns separate. In the first three columns it would really be more correct to multiply the figures by $M_0$, $P_0$, and

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$Q_0$ respectively; but it saves repeating these letters, if it is remembered that the figures in these columns are really factors to $M_0$, $P_0$, and $Q_0$ respectively. The factors of $P_0$ are the ordinates in feet of the points considered, and those of $Q_0$ the abscissæ. Strictly speaking, all these moments ought to be taken about the neutral axes of the sections at the various points; but, as these axes will practically pass through the moulding edge of

TABLE II.

| No. of Section. | Moment of Inertia I. | $\frac{M}{I}$ | $\frac{Py}{I}$ | $\frac{Qx}{I}$ | $\frac{S}{I}$ | $\frac{C}{I}$ | $\frac{H+V}{I}$ |
|---|---|---|---|---|---|---|---|
| 0 | 135·0 | + ·007 | + ·00 | − ·00 | + 0 | + 0 | − 0 |
| 1 | 95·5 | + ·010 | + ·01 | − ·02 | + 2 | + 11 | − 17 |
| 2 | 65·0 | + ·015 | + ·02 | − ·06 | + 11 | + 65 | − 97 |
| 3 | 49·5 | + ·020 | + ·04 | − ·13 | + 31 | + 185 | − 273 |
| 4 | 13·2 | + ·076 | + ·21 | − ·66 | + 198 | + 1,166 | − 1,795 |
| 5 | 4·8 | + ·208 | + ·97 | − 2·07 | + 770 | + 4,041 | − 6,810 |
| 6 | 1·4 | + ·714 | + 4·96 | − 7·14 | + 2,643 | + 13,860 | − 27,640 |
| 7 | 1·4 | + ·714 | + 6·61 | − 7·14 | + 2,643 | + 13,860 | − 32,350 |
| 8 | 1·6 | + ·625 | + 7·24 | − 6·19 | + 2,312 | + 11,870 | − 32,190 |
| | | | | | | | |
| 8 | 4·1 | + ·244 | + 2·82 | − 2·41 | + 902 | + 4,634 | − 12,560 |
| 9 | 2·0 | + ·500 | + 5·87 | − 3·72 | + 517 | + 5,040 | − 19,650 |
| 10 | 2·0 | + ·500 | + 5·92 | − 2·47 | − 955 | + 495 | − 13,395 |
| 11 | 2·0 | + ·500 | + 5·99 | − 1·24 | − 2,550 | − 4,025 | − 7,150 |
| 12 | 2·0 | + ·500 | + 6·00 | − ·00 | − 4,300 | − 8,600 | − 700 |

the frames and beams, the latter line has been assumed to represent the curve through the centre of gravity of the sections. In cases where the floors are deep, or where double bottoms are fitted, and the centres of gravity of the sections consequently at some distance from the shell plating of the vessel, it may be necessary to work from a correct curve through the various centres of gravity or neutral axes.

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The moments of the weight of the structure and cargo may be calculated in the usual way, and the moment of the vertical pressure of the water will be the moment of the displacement of the section from 0 up to the point considered. For points in

TABLE III.

| No. of Section. | Simpson's Multipliers. | $\int \frac{M}{I} d\,l.$ | $\int \frac{P\,y}{I} d\,l.$ | $\int \frac{Q\,x}{I} d\,l.$ | $\int \frac{S}{I} d\,l.$ | $\int \frac{C}{I} d\,l.$ | $\int \frac{(H+V)}{I} d\,l.$ |
|---|---|---|---|---|---|---|---|
| 0 | 1 | + ·007 | + ·00 | − ·00 | + 0 | + 0 | − 0 |
| 1 | 4 | + ·040 | + ·04 | − ·08 | + 8 | + 44 | − 68 |
| 2 | 2 | + ·030 | + ·04 | − ·12 | + 22 | + 130 | − 194 |
| 3 | 4 | + ·080 | + ·16 | − ·52 | + 124 | + 740 | − 1,092 |
| 4 | 2 | + ·152 | + ·42 | − 1·32 | + 396 | + 2,332 | − 3,590 |
| 5 | 4 | + ·832 | + 3·88 | − 8·28 | + 3,080 | + 16,164 | − 27,240 |
| 6 | 2 | + 1·428 | + 9·92 | − 14·28 | + 5,286 | + 27,720 | − 55,280 |
| 7 | 4 | + 2·856 | + 26·44 | − 28·56 | + 10,572 | + 55,440 | − 129,400 |
| 8 | 1 | + ·625 | + 7·24 | − 6·19 | + 2,312 | + 11,870 | − 32,190 |
| | | + 6·050 | + 48·14 | − 59·35 | + 21,800 | + 114,440 | − 249,054 |
| 8 | 1 | + ·244 | + 2·82 | − 2·41 | + 902 | + 4,634 | − 12,560 |
| 9 | 4 | + 2·000 | + 23·48 | − 14·88 | + 2,068 | + 20,160 | − 78,600 |
| 10 | 2 | + 1·000 | + 11·84 | − 4·94 | − 1,910 | + 990 | − 26,790 |
| 11 | 4 | + 2·000 | + 23·96 | − 4·96 | − 10,200 | − 16,100 | − 28,600 |
| 12 | 1 | + ·500 | + 6·00 | − ·00 | − 4,300 | − 8,600 | − 700 |
| | | + 5·744 | + 68·10 | − 27·19 | − 13,440 | + 1,084 | − 147,250 |
| | | | | | | | $\times \frac{2·475}{2·310}$ |

$$+ 6·154 \quad + 72·93 \quad - 29·13 \quad - 14,400 + 1,160 - 157,750$$

$$+ 6·050 \quad + 48·14 \quad - 59·35 \quad + 21,800 + 114,440 - 249,054$$

$$+ 12·204\,M_0 + 121·07\,P_0 - 88·48\,Q_0 + 7·400 + 115,600 - 406,804 = 0. \qquad (10)$$

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the flat of the side of the section, and for points above, the moments will, therefore, be equal to the total displacement of the half section, multiplied by the horizontal distance between the point considered and the centre of buoyancy of the half section.

TABLE IV.

| No. of Section. | $y$ | $\int \frac{M}{I} y\,dl.$ | $\int \frac{P}{I} y^2\,dl.$ | $\int \frac{Qx}{I} y\,dl.$ | $\int \frac{S}{I} y\,dl.$ | $\int \frac{C}{I} y\,dl.$ | $\int \frac{(H+V)}{I} y\,dl.$ |
|---|---|---|---|---|---|---|---|
| 0 | ·00 | + ·000 | + ·00 | − ·00 | + 0 | + 0 | − 0 |
| 1 | ·60 | + ·024 | + ·02 | − ·05 | + 5 | + 26 | − 41 |
| 2 | 1·20 | + ·036 | + ·05 | − ·14 | + 26 | + 156 | − 233 |
| 3 | 1·80 | + ·144 | + ·29 | − ·94 | + 223 | + 1,332 | − 1,965 |
| 4 | 2·75 | + ·418 | + 1·16 | − 3·63 | + 1,089 | + 6,413 | − 9,872 |
| 5 | 4·65 | + 3·868 | + 18·04 | − 38·50 | + 14,320 | + 75,160 | − 126,660 |
| 6 | 6·95 | + 9·924 | + 68·94 | − 99·24 | + 36,740 | + 192,600 | − 384,200 |
| 7 | 9·25 | + 26·420 | + 244·60 | − 264·20 | + 97,800 | + 512,800 | − 1,197,000 |
| 8 | 11·58 | + 7·237 | + 83·84 | − 71·68 | + 26,770 | + 137,450 | − 372,800 |
| | | + 48·071 | + 416·94 | − 478·38 | + 176,973 | + 925,937 | − 2,092,771 |
| 8 | 11·58 | + 2·825 | + 32·65 | − 27·90 | + 10,450 | + 53,660 | − 145,400 |
| 9 | 11·75 | + 23·500 | + 275·86 | − 174·83 | + 24,300 | + 236,900 | − 923,500 |
| 10 | 11·85 | + 11·850 | + 140·30 | − 58·54 | − 22,630 | + 11,730 | − 317,400 |
| 11 | 11·98 | + 23·960 | + 287·05 | − 59·42 | − 122,200 | − 192,900 | − 342,600 |
| 12 | 12·00 | + 6·000 | + 72·00 | − ·00 | − 51,600 | − 103,200 | − 8,400 |
| | | + 68·135 | + 807·86 | − 320·69 | − 161,680 | + 6,190 | − 1,737,300 |
| | | | | | | | $\times \frac{2·475}{2·310}$ |

$$72·990 + 865·50 - 343·60 - 173{,}200 + 6{,}630 - 1{,}861{,}300$$

$$48·071 + 416·94 - 478·38 + 176{,}973 + 925{,}937 - 2{,}092{,}771$$

$$121·061\,M_0 + 1282·44\,P_0 - 821·98\,Q_0 + 3{,}773 + 932{,}567 - 3{,}954{,}071 = 0. \qquad (11)$$

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In the calculation of the moments of the horizontal pressures, it may be convenient to construct a curve of varying intensities, similar to that provided by the outline of the section itself, in the case of the vertical pressures. The curve will in still water

TABLE V.

| No. of Section. | $x$ | $\int \frac{M}{I} x\,d\,l.$ | $\int \frac{P\,y}{I} x\,d\,l.$ | $\int \frac{Q}{I} x^2\,d\,l.$ | $\int \frac{S}{I} x\,d\,l.$ | $\int \frac{C}{I} x\,d\,l.$ | $\int \frac{(H+V)}{I} x\,d\,l.$ |
|---|---|---|---|---|---|---|---|
| 0 | ·0 | + ·000 | + ·00 | − ·00 | + 0 | + 0 | − 0 |
| 1 | 2·22 | + ·088 | + ·08 | − ·18 | + 18 | + 98 | − 151 |
| 2 | 4·44 | + ·133 | + .18 | − ·53 | + 98 | + 577 | − 861 |
| 3 | 6·66 | + ·533 | + 1·06 | − 3·46 | + 826 | + 4,930 | − 7,270 |
| 4 | 8.75 | + 1·330 | + 3·67 | − 11·55 | + 3,465 | + 20,400 | − 31,400 |
| 5 | 9·95 | + 8·278 | + 38·60 | − 82·38 | + 30,640 | + 160,800 | − 271,000 |
| 6 | 10·00 | + 14·280 | + 99·20 | − 142·80 | + 52,860 | + 277,200 | − 552,800 |
| 7 | 10·00 | + 28·560 | + 264·40 | − 285·60 | + 105,720 | + 554,400 | −1,294,000 |
| 8 | 9·90 | + 6·187 | + 71·67 | − 61·27 | + 22,890 | + 117,500 | − 318,700 |
| | | + 59·389 | + 478·86 | − 587·77 | + 216,517 | +1,135,905 | −2,476,182 |
| 8 | 9·90 | + 2·415 | + 27·92 | − 23·86 | + 8,930 | + 45,870 | − 124,300 |
| 9 | 7·45 | + 14·900 | + 174·92 | − 110·85 | + 15,400 | + 150,200 | − 585,500 |
| 10 | 4·95 | + 4·950 | + 58·61 | − 24·45 | − 9,450 | + 4,900 | − 132,600 |
| 11 | 2·48 | + 4·960 | + 59·42 | − 12·30 | − 25,300 | − 39,900 | − 70,900 |
| 12 | ·00 | + ·000 | + ·00 | − ·00 | − 0 | − 0 | − 0 |
| | | + 27·225 | + 320·97 | − 171·46 | − 10,420 | + 156,170 | − 913,300 |
| | | | | | | | $\times \frac{2·475}{2·310}$ |

29·274 + 343·90 − 183·70 − 11,160 + 167,300 − 978,500

59·389 + 478·86 − 587·77 + 216,517 + 1,135,905 − 2,476,182

$$88·663\ M_0 + 822·76\ P_0 - 771·47\ Q_0 + 205,357 + 1,303,205 - 3,454,682 = 0. \qquad (12)$$



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be a straight line from the water-line downwards, and making an angle of 45° with the vertical, as shown on Fig. 2 (Plate LIII.). If horizontal lines are drawn across the triangle at the level of the various points on the section, then the moments of the horizontal pressures will be equal to the moments of the trapezoids bounded by the base line, and the horizontal line through the points considered, and can be readily obtained when the area and centres of gravity of the trapezoids are determined. For points above the water-line the moments will be the product of the total area of the triangle and the vertical distance between its centre of gravity and the point considered.

It is essential that the direction, or sign, of the moments given in Table I. (page 277) should be attended to. The horizontal forces have been assumed to be positive, if acting towards the right of the sketch, and the vertical ones positive, if acting upwards. The moments have been taken to be positive, if acting contrary to the hands of a watch. The sum of the figures across all the columns for a given point represents, therefore, the total bending moment at that place, both in magnitude and direction.

Next Table II. (page 278) is prepared. The first column gives the moments of inertia at the various sections in inch$^4$ units, and the figures in the following six columns represent the corresponding figures in Table I., divided by the moments of inertia, the columns being headed accordingly.

Table III. (page 279) represents the integration, by means of Simpson's rule, of the various columns in Table II., the operation being performed separately for the frame and beam parts of the girder; and, instead of multiplying both results by the corresponding interval, the results for the beam part are multiplied by the interval of the part from 8 to 12, and divided by the interval of the part from 0 to 8. By adding the two results and equating to zero we get equation (10).

Table IV. (page 280) represents the integrations of $\frac{M}{I} y\, d\, l$, and is obtained from Table III. by multiplying the numbers by the corresponding values of the ordinates of the points, as given in the column following the numbers of the sections. The results are added up and equated to zero, as before, and equation (11) is obtained.

Table V. (page 281) represents the integration of $\frac{M}{I} x\, d\, l$, and is derived from Table III. by multiplying all the numbers by the corresponding abscissæ of the sections, as tabulated in the second column of Table V. The results are summed up, and equation (12) is obtained.

These three equations are then solved, and the values of the unknowns are found to be as follows:—

$$P_0 = +\ 2{,}669 \text{ lbs.}$$
$$Q_0 = -\ \ 272 \text{ lbs.}$$
$$M_0 = -\ 5{,}197 \text{ ft. lbs.}$$

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By substituting these values for $P_0$, $Q_0$, and $M_0$ in Table I., and adding the figures for each point, we get the total bending moments over the whole of the girder. These are recorded in Table VI. below, as well as the corresponding moments of resistance of the sections, and the maximum stress on the material in tons per square inch.

The problem of determining the transverse stresses under given conditions is thus solved, at least in the case of a single-deck vessel, having the weight of the cargo and

TABLE VI.

| No. of Section. | Bending Moments in Foot Lbs. | Moments of Resistance in Inch³. | Stress in Tons per Square Inch. |
|---|---|---|---|
| 0 | – 5,197 | 14·40 | 1·9 |
| 1 | – 3,360 | 11·70 | 1·5 |
| 2 | – 2,191 | 9·15 | 1·3 |
| 3 | – 1,451 | 7·75 | 1·0 |
| 4 | – 1,157 | 3·55 | 1·8 |
| 5 | + 319 | 1·80 | ·9 |
| 6 | + 473 | ·48 | 5·3 |
| 7 | + 13 | ·48 | ·1 |
| 8 | – 394 | ·56 | 3·8 |
| 8 | – 394 | ·86 | 2·5 |
| 9 | + 4 | ·60 | ·0 |
| 10 | + 44 | ·60 | ·4 |
| 11 | – 2 | ·60 | ·0 |
| 12 | – 367 | ·60 | 3·3 |

structure equal to the buoyancy at the point considered. The process is simple and direct, and can easily be worked to without the knowledge of any more integral calculus than is required to understand the working of Simpson's rules. The method of considering certain forces or moments as known, and treating them as such, until they can be found by the principle of least work, can be modified to suit circumstances, not

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only in connection with the transverse strength, but also those of a great number of other cases, where the forces are indeterminable by the ordinary statical methods.

To revert to the case under consideration, a curve K G L has been drawn on Fig. 2 (Plate LIII.), representing the bending moments. The ordinates are set off from the corresponding points normally to the section and to that side of the girder which would be in tension. In this way the curve represents also to a certain extent the form the strained section would tend to assume. An examination of the curve and of the moments of the structure, cargo, and water pressures will show that the horizontal pressure of the water on the sides of the vessel is the most important factor in determining the magnitude of the stresses, because, although the pressure of the water on the bottom is somewhat in excess of that of the weight of the structure and cargo, and the floors might, therefore, be expected to bend inwards, the pressure on the sides is sufficient to completely reverse this bending tendency, so that the largest bending moment on the girder is at the centre of the floors and is tending to bend the floors outwards. The greatest stress, it will be seen from Table VI. (page 283), is at the middle of the side where the moment of resistance is least. It would, however, have been at the bilge, if the floor, and reversed frames had been stopped at a less height from the base line.

It was assumed in the above calculations that the weight of the structure and cargo of the section considered was equal to the displacement, and that the section was independent of the neighbouring material. This may not be the case, and very often is far from the case, as some holds in a vessel may be fully loaded, and others empty or nearly so. In way of the empty holds the buoyancy will, therefore, be in excess of the weights. The most severe cases will usually occur when the vessel is loaded down with water ballast concentrated in deep tanks, while other holds may be completely empty. At the latter places the water pressure on the bottom will then tend to force the floors inwards. Let the extreme case be assumed, when all the cargo is removed, in the case shown in Fig. 2 (Plate LIII.), the draught remaining the same. The ring assumed cut out of the vessel cannot, therefore, be in equilibrium without forces from the neighbouring parts of the structure. The unequal distribution of the weight and buoyancy must be balanced by shearing forces.

There may be very large shearing forces at both ends of the one foot length of the vessel which we are considering, but it is only the difference between these forces which affects the transverse bending moments, the remainder of the forces simply pass through from one side to the other. In order that the athwartships section, or ring, may be in equilibrium, it is necessary that this difference in the shearing forces should be equal to the excess of buoyancy or weight. When all the cargo is removed and the draught remains the same, then this force must be equal to the weight of

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the cargo, or, in the present instance, 3,670 lbs. It will be practically only distributed over the vertical part of the side plating of the vessel, and there will be no appreciable error in assuming it to be entirely confined to the parts of the girder lying between 5 and 8 (Fig. 2, Plate LIII.).

To calculate the stresses for the altered conditions with an excess of buoyancy of 3,670 lbs. over the weight, we have only to introduce in Tables I., II., III., IV., and V. the moments of the shearing force applied to the shell plating between sections 5 and 8, and to omit the moments of the cargo given in the columns marked C. The moments of the shearing force will be zero from 0 to 8, and will have negative values for the beam part of the girder, varying as the distance of the points considered from the side of the vessel. We may then find the equations as before, substitute the values found in Table I. with the necessary alterations in column C, and the bending moments are determined. The curve $K_1$ $G_1$ $L_1$ (Fig. 2) represents these moments. It will be noticed that the excess of buoyancy at the bottom is now so large that the direction of the bending moments on the floors is reversed.

A few cases have been worked out, showing the modification in the stresses due to variations in the form of the vessel. Thus, Fig. 3 (Plate LIII.) shows the case of a vessel of the same dimensions, draught, and scantlings, but of a somewhat fuller form than that shown on Fig. 2. The bending moments indicated by the curve $K_1$ $G_1$ $L_1$ are for the vessel floating light, and those shown by K G L for the loaded condition. The importance of the side pressure is again proved here, as it completely reverses the largest bending moments, although the excess of vertical pressure over weight at the bottom is the same in both cases.

Fig. 4 shows a vessel of the same dimensions, draught, and scantlings, but of a much finer form than the previous two. The curve K G L again represents the case of the loaded vessel. In comparing the three vessels of varying fineness of midship section, it will be seen that the bending moments on the framing are considerably reduced in the finer vessels, but that the moments at the centre of the floors are on the other hand very much increased. One conclusion drawn from this fact would be that the frames of the finer vessels might be of somewhat less scantlings than in full vessels, provided the strength at the floors is sufficient.

The curve $K_1$ $G_1$ $L_1$ (Fig. 4) gives the bending moment for the case where lead ballast, stored between the floors, has been substituted for the cargo. The bending is in this case very much increased at the middle of the floors, as might be expected.

Figs. 5 and 6 (Plate LIII.) represent the cases of two racing yachts of different types, and heavily ballasted with lead. In these two cases the starting point for the integration is not taken at the centre of the vessel, but at the top of the floors at the

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frame edge, the part below being taken as rigid. The effect of any pillars fitted at the centre has also been neglected. The vertical force at the point O is therefore equal to the difference between the weight and buoyancy over the part between O and the middle line of the vessel. There are thus only two unknowns to be found, and only two equations are required.

It will be noticed that in all the above calculations the cargo has been assumed to exert only a vertical pressure on the floors and frames. Whether this assumption is correct or not depends on the nature of the cargo. In the case of timber, or rails, the assumption will be practically true. If, on the other hand, the cargo consists of liquid, then it will exert a horizontal pressure similar to that of the water on the exterior of the vessel. The amount of such side pressure will depend on the viscosity of the liquid. The nature of the cargo being known, the calculations can be made accordingly. Concentrated weights are also easily included in the calculations.

If the ship is rolling, and the conditions given, the reactions of the weights and their moments may be found and included in the estimate of the bending tendencies. In that case, the symmetry of the structure and the applied forces is, however, destroyed, and it is impossible to deal with one side only. The integration must, in that case, be performed over the whole of the section, which will make the solution more complicated, but not impracticable.

The vessels dealt with so far have been assumed to be all single-decked. If two decks, or tiers of beams, are fitted as shown in Fig. 7 (Plate LIII.), it will be found that three unknowns are not sufficient to determine the case. The three unknown quantities at the keel of the vessel will, as before, be sufficient to determine all the moments up to the lower deck beams, but not further. It becomes necessary to assume a new set of three unknowns for the centre of the lower deck beams. The case is then the same as before, only we are dealing with six unknowns, $M_0$, $P_0$, $Q_0$, $M'$, $P'$, and $Q'$, and have to get six equations for their determination. The first three unknowns only enter into the integration over $K G_1$, the second three only over the part $L_1 G_1$, and the whole six over $G_1 G L$, but the whole of the results must be summed up to obtain the equations.

Again, if rows of pillars are introduced between the centre and the side of the vessel, it is necessary to assume one unknown force for each row. It is only necessary to assume a direct force, because a pillar cannot transmit any appreciable shearing force, or bending moment. The process of calculation becomes more lengthy with the introduction of so many unknowns, and it may be desirable to adopt some modified method. The vessels having several rows of pillars and several tiers of beams have in most cases a double bottom fitted, which possesses sufficient transverse strength, and may, in the calculation of the strength of the frames, be taken as practically rigid.

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The frames may also be supposed to be held rigidly, fixed at the lowest complete tier of beams, in particular when the beams are loaded with cargo, and, therefore, tending to bend the frames in a direction contrary to that due to the pressure of the water. The estimate of the strength of the frames of a ship with more tiers of beams and rows of pillars may, therefore, in the cases of most practical importance, be done on the assumption that the frame girders are fixed at the floor ends and at the lowest complete tier of beams. It is, then, only necessary to assume two unknowns in this case, say, a horizontal force $P_0$ at O (Fig. 10, Plate LIII.), and a bending moment $M_0$. The method of finding $P_0$ and $M_0$ is the same as previously explained in the case of three unknowns. The integration is only performed for the part of the frames between floor heads and lower deck beams, and only the forces acting on this part are included in the calculation.

### Stringers, Bulkheads, and Wide-spaced Beams.

The considerations of the transverse strength of the girders, shown in Figs. 1 to 6 (Plate LIII.), have been based on the assumption that the ring sections were independent of the neighbouring parts of the structure, except in the case of the shearing forces due to unequal distribution of weight and buoyancy. Other forces may, however, be transmitted to the transverse girders, and either tend to support them or to further strain them.

It is clear that in the flat of the vessel amidships no possible longitudinal either pull or push stresses can appreciably affect the transverse girders, either by giving them support or by straining them: the duty of the frames, floors, and beams being in this respect only one of keeping the longitudinal plating in position, and preventing it crumpling under compressive stresses. If, on the other hand, there is considerable curvature in the water-lines of the vessel, as in small vessels, or towards the ends in larger and full vessels, then some support may be given to the frames by the shell plating acting in the way of an arch, if the ends of the curved parts are rigidly supported. The assistance thus lent to the transverse strength must, however, be small, except in small boats with comparatively thick planking or plating.

The effect of twisting moments must also be inappreciable on the transverse girders.

Transverse bulkheads are very important factors in the athwartship strength of ships. Their effect must be transmitted to the frames and beams through the shell plating, deck plating, and stringers.

In dealing first with the effect transmitted through the deck and shell plating, we may assume that transmitted through the stringers to be *nil*. If the system of the straining forces is symmetrical, as in the case of a vessel floating freely in still water,

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then the bulkheads will do very little more work than one of the ordinary transverse girders. If, however, the vessel is rolling, or otherwise subjected to unsymmetrical straining agencies, then the deck may tend to move relatively to the bottom, or the one side relatively to the other. It is this motion which the bulkheads are effective in preventing, and as they are practically rigid, compared with the ordinary transverse girders, it follows that they will relieve the framing of almost the entire amount of straining due to the above causes. The nature of the support afforded by the bulkheads may be illustrated by the sketch, Fig. 8 (Plate LIII.). The support is equivalent to providing a fixed point in both sides and in the deck and bottom plating. We may, therefore, assume an absolutely rigid frame construction, fixed in the vessel and holding pistons capable of sliding in circular holes, and attached to the frames, beams, and floors of the vessel, as indicated on Fig. 8, by flexible joints. The vertical pistons must be assumed connected where pillars are tying the deck to the floors. The support of the bulkheads is, therefore, equivalent to providing certain forces in the direction of the plating at the joints of the imaginary pistons. These forces could be determined by including them as additional unknowns in the method of least work, but as the bulkheads are almost entirely preventing the racking deformation of the transverse sections, we may assume the frames, floors, &c., to take only the direct local forces, and calculate the stresses accordingly. The racking forces, moreover, cannot be very great, compared with other straining agencies, as is evident from the fact that sailing vessels, in which they would be largest, have usually no bulkheads, and need practically no larger framing on that account.

The bulkheads will be effective in preventing racking as long as their spacing is close enough to prevent the deck or shell plating bending appreciably in the direction of the plating, which will be the case in nearly all ordinary steamers.

The other way in which the bulkheads support the framing is through the hold stringers. The frames and stringers between the lowest tier of beams and the floor heads have, together, to support the pressure of the water, and all other forces which may act on the inside of the framing. Let, for simplicity's sake, the inside forces be neglected, and let the deflexion caused in the ship's side by the water pressure be $\Delta$, we then have—

$$\Delta = \mu \frac{l^3}{I} \rho = \mu \frac{l_1^3}{I_1} \rho_1,$$

where $l$ and $l_1$ are the depth and length of the hold respectively, I and $I_1$ the moments of inertia respectively of the total number of frame sections, and of the total number of stringers. $\rho$ and $\rho_1$ are the pressures borne respectively by the frames and stringers and $\mu$ is a constant.

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We further have—

$$\frac{\rho}{\rho_1} = \frac{l_1^3 \, I}{l^3 \, I_1},$$

and $\rho + \rho_1$ is equal to the total pressure, and is therefore known. The pressure taken by each of the two sets of girders can therefore be determined, and the bending moments and stresses found. The method is not strictly correct, but is probably practically so when $l_1$ is much larger than $l$, which is nearly always the case, except in vessels carrying liquids in bulk. When $l_1$ approaches $l$ in length, the actual strength will probably be in excess of that calculated by the above method. It will be noticed that the length of the hold is a very important factor in determining the amount of support lent by the bulkheads to the framing.

Strong hold beams being practically rigid, as far as compressive forces are concerned, may be dealt with similarly to bulkheads, as regards the support given to the framing. $l_1$ is then the spacing between the beams, and I the moment of inertia of the corresponding number of frames and reversed frames, including the shell plating. $l$ is the distance, as before, between the floor ends and the lowest complete tier of beams. $I_1$ is the moment of inertia of the longitudinal material, and is usually made up almost entirely of the deep stringers fitted at the ends of wide-spaced hold beams.

The ratio of the pressures transmitted through the frame girders and through the hold stringers are given in a few cases in the table below. The vessels are constructed in the ordinary way with either wide-spaced hold beams, web frames, or deep framing.

| Dimensions of Vessel. | Description. | Length of Holds. | $\frac{\rho}{\rho_1} = \frac{l_1^3 \, I}{l^3 \, I_1}$ | | |
|---|---|---|---|---|---|
| | | | Wide-spaced Beams. | Web Frames. | Deep Framing. |
| 256 by 38·2 by 20·6 feet mld. | One deck | 90 feet | ·08 | 720 | 132 |
| 286 by 42·7 by 23·0 „ „ | „ „ | 68 „ | ·05 | 107 | 24 |
| 340 by 50·7 by 27·3 „ „ | Two decks | 81 „ | — | 720 | 172 |

It will be observed, that the work done by the stringers in the cases of deep framing and web frames is very small compared to that done by the frames and web frames. In the latter case the stringers are, however, necessary to support the shallow frames between the web frames. In the cases of wide-spaced beams the work done by the deep stringers in transmitting the pressures to the points of support is large, and



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the frames have practically only to take the forces between the stringer and the deck or floors; or, in other words, the stringer is nearly a rigid point of support for the frames. In estimating the bending moments on the frames, we have therefore two parts to deal with, one due to the bending over the lengths $l$, or between the floors and lowest deck, in which case only a small percentage of the forces is taken by the frames: in the case of the 256 ft. vessel in the above table, it is about 8 per cent., and in the case of the 286 ft. vessel, about 5 per cent. In calculating the other part of the bending moments we must take the remaining amounts of the forces, in the above-mentioned vessels 92 per cent. and 95 per cent. respectively; but the lengths between supports is reduced to the distance between the deep stringer and the floors or the deck above, according to which part is being considered.

In the deep framing and web frame vessels the bending of the frames between the hold stringers is usually of little importance compared with the yielding between the floors and the lowest deck, and it will not, as a rule, be necessary to make any allowance for this kind of straining, in particular as the greatest stress in the case of the web frame vessels comes on the inside of the webs, and not on the shallow frames.

The division of the work of supporting the applied forces being found, the stresses on the stringers can be calculated by the ordinary methods, treating the bulkheads and strong beams as points of support. The stresses on the framing can be estimated by the method of least work, taking of all the applied forces, water pressure, weight, or rolling reactions, the proper proportion left over by the stringers.

At the ends of vessels where the stringers have some horizontal curvature they may be slightly more efficient as supports to the frames, but their efficiency is usually again destroyed by the vertical curvature due to the sheer of the vessel.

### Stresses Due to Shocks from Waves.

The pressure of the water on the sides of the vessel and on the bottom has, so far, been assumed to be that due to the steady pressure of the water when at rest. The stresses will necessarily be greater when a ship is at sea rolling and subjected to blows from waves. The horizontal pressure of the water being by far the most important force acting on the frames of a vessel, it is desirable that we should know, as far as possible, what it will be under the worse conditions met with at sea. Unfortunately, there appears to be very little direct information as to the striking force of a wave. Some experiments were made by Mr. Thomas Stevenson, and the maximum striking force, as measured by dynamometers, was found by him to be 3 to 3½ tons per square foot for Atlantic waves striking a vertical wall or rock. The average forces were, however, much smaller, being 611 lbs. and 2,806 lbs. per square foot in the summer and winter months respectively. As regards the maximum values, it will be evident that

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they cannot be anything like so large where the waves are striking a floating vessel, which will move with the striking body of water, if this is of large dimensions.

The distribution of the maximum values of the striking forces against a sea wall, as recorded by Mr. Stevenson in his book on lighthouse construction, is as indicated on Fig. 9 (Plate LIII.), the greatest forces occurring at the still water level. The forces of waves striking a ship's side will probably be distributed somewhat in the same manner, and they must be added to the steady pressure due to head of water. The additional pressure will be greatest at the water-line, and from there taper to zero at some distance above and below. Instead of the line T W (Fig. 10), representing the horizontal pressures, we shall then get a curve T L N. The exact character of this curve could be determined by dynamometric measurements, and, considering the ease with which such determinations could be made by a comparatively simple apparatus, it appears peculiar that so little exact information exists on this subject.

It may be reasonably assumed that there is no variation in the pressure at the lowest part of the vessel, and the curve will therefore start from T. Let the form be that of a common parabola, touching W T at T, and the vertical through W at a distance above W equal to K W, or the draught of the vessel. W L will then be equal to $\frac{1}{4}$ K W, or $\frac{1}{4}$ K T, and G N will for ordinary seagoing vessels with the usual freeboard be about 3 ft. As no exact scientific determination of these pressures is available, it may be desirable to compare the results given by the above curve with such information as exists on the subject, and as can be derived from the efficiency or otherwise of actual structures.

From the knowledge of rudder stocks being broken by striking seas, the force of the waves can be roughly estimated by eliminating any pressure due to the speed of the vessel, and it is found to be about ·2 of a ton per square foot. Assuming this to be the value at the load water-line, it will correspond to W L (Fig. 10, Plate LIII.), being 7 ft., which will be the case when the draught is 28 ft., on the above assumption as to the curve being a parabola. Another verification of this kind may be found in the fact that the pressure on the weather deck beams required to give the same stress or factor of safety, as in the case of the lower deck beams, agrees very closely with the pressure G N at the gunwale, as assumed by the above method.

It would, therefore, appear that the assumptions as to the variation in the pressures on the vessel's sides are fairly correct, and might be adopted for purposes of comparison until more exact information is provided.

Below is given a table of the stresses on the framing of various vessels worked out on this basis. The vessels are of the usual types, with wide-spaced hold beams, web frames, or deep framing, and are of the scantlings required for the highest

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class in Lloyd's Register. The bending moments are worked out as explained above, the work of supporting the water pressures being divided between the frames and the stringers, and the method of least work employed in determining the bending moments on the frames. The work in this case is comparatively simple, owing to only two unknowns being necessary, and the moment of inertia being uniform for the entire length of framing considered. The curve of bending moments is shown on Fig. 10 (Plate LIII.), for the case of the 340 ft. vessel with deep framing.

| Dimensions of Vessel. | Description. | Stresses in Tons per square inch on Inside of Reversed Frames or Webs. | | | |
|---|---|---|---|---|---|
| | | Ordinary Frames. | Hold Beams. | Web Frames. | Deep Framing. |
| 256 by 38·2 by 20·6 feet moulded ... ... | One deck | — | 4·6 | 6·7 | 6·1 |
| 286 by 42·7 by 23·0 „ „ ... ... | „ „ | — | 5·7 | 6·9 | 6·7 |
| 340 by 50·7 by 27·3 „ „ ... ... | Two decks | — | — | 4·9 | 4·3 |
| 382 by 50·7 by 30·8 „ „ ... ... | „ „ | — | 5·1 | 7·5 | 6·2 |
| 430 by 57·0 by 34·7 „ „ ... ... | Three decks | 6·4 | — | — | — |

The stresses given in the above table are calculated for the midships part of the vessel, and the stresses at the ends will be somewhat higher, owing to the greater depth in ships having sheer.

The two single-deck vessels have been dealt with similarly to the two-deck ones, the integration being performed only for the part of the frames between the bilge and the gunwale. The error in thus dealing with a single-deck vessel is not great when the ship is of full form and fully loaded, as may be seen by comparing the curves of bending moments shown on Figs 3 and 10 (Plate LIII.).

I regret that I have had to confine myself mainly to indicating methods, and that want of time and my ordinary duties have prevented me from giving more results of the proposed methods of calculation, particularly in the cases of vessels having several tiers of beams, and where several sets of unknowns would have to be used in the complete solution.

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## DISCUSSION.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Ladies and Gentlemen, we have had a highly interesting paper from Mr. Bruhn, and I only regret that he has had to cut out so much of it on account of the shortness of time at our disposal.

Mr. BENJAMIN MARTELL (Vice-President): My Lord, Ladies, and Gentlemen, I consider this paper to be most valuable. The question of the longitudinal stresses brought on a ship has been under consideration and has been investigated for a very long time, and most useful deductions have been made with that object in view; but the question of transverse stresses on ships has not been considered to the extent that it might have been, and that induced my assistants, Mr. Jenkins and the late Mr. Reed, to undertake an investigation of this subject, and they read a paper before the Institution of Naval Architects that was highly thought of and very much appreciated by the members. Mr. Bruhn subsequently took up the consideration of this question, and one great advantage to be derived from the papers that he reads is that his scientific investigations always tend to practical results, enabling the designers and constructors of the ships to grasp the importance of the question raised and to deal with the requirements embodied in his papers. The Council of the Institution were so well aware of this fact that they awarded Mr. Bruhn the Gold Medal of the Institution. The subjects that Mr. Bruhn has gone into are of a very important character, and it is a pity that he has had to eliminate so much of his paper, not having time to go into it fully. But it is a paper that will well bear reading and studying by naval architects. It opens out a new light on many points with regard to the transverse strength of ships, and I can only hope, for his own reputation's sake, and likewise for the advantage of Lloyd's Register, and all the members of this Institution, that Mr. Bruhn will continue to give his attention to important subjects of this class, which interest us so much. I do not think I need analyse anything that is said here. It is all of such a plain character, that anybody with some knowledge of mathematics may study it, and derive great benefit from it. I would therefore suggest that Mr. Bruhn be assured that the members of this Institution are highly gratified with the excellent paper that he has placed before them.

Mr. FRANCIS ELGAR, LL.D., F.R.S. (Vice-President): My Lord, Ladies, and Gentlemen, I think the reader of this paper is a very bold man, in attacking one of the most difficult problems that are to be found in connection with the calculation of the strength of structures. He has very much understated, in my judgment, the difficulty of the problem, in saying, on the first page of the paper, that "the case is somewhat more complicated than that of estimating the longitudinal strength." I should say it is not only much more complicated, but, without having had time to examine closely Mr. Bruhn's mathematical investigations, I would go so far as to add that, notwithstanding those investigations, it remains not simply in the condition of being a very difficult problem, but that it is an indeterminate one. I have only seen the paper during the last few minutes, and cannot therefore pretend to do full justice to it. But it seems to me that in attacking the problem as Mr. Bruhn does, by taking the forces into account that act upon the structure, and also employing the principle of least work, he must still be left with the necessity for making some further assumption in order to arrive at a complete solution. I do not at present see—I have not had time to discover—what that assumption is. But, however that may be, I am satisfied that Mr. Bruhn has made a distinct advance on former attempts to solve the problem by his use of the principle of least work. There is no doubt that that is the best principle to go upon, and that its introduction has enabled Mr. Bruhn to

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improve upon the excellent work done in connection with this difficult problem by his predecessors. Whether he has succeeded in arriving at a complete solution of what has been, up to the present, an indeterminate problem, is a much larger question, and one which, notwithstanding a very superficial reading of the paper, I should answer in the negative. Mr. Bruhn may be able to tell us, perhaps, in his reply, whether I am right in supposing that there must be some other assumption adopted by him to enable him to arrive at a solution of his equations. I quite agree with Mr. Bruhn in considering that the principal elements of transverse structural strength, and the chief factors in preserving the transverse form of a ship, are the transverse bulkheads. In the large ships now under construction, which have a good number of transverse bulkheads, these bulkheads add to the efficiency of the ship, not only as an element of safety in the event of compartments being flooded, but also as a principal element of transverse structural strength. I would, however, not go so far as Mr. Bruhn in thinking that the number of bulkheads we are accustomed to are going to relieve the framing of almost the entire amount of straining, except what might be due to the action of local forces. That is an assumption I do not follow. I think that in any case there must be a very great deal of transverse structural straining brought upon the framing of a ship, which cannot be taken up by the bulkheads; and, if that be so, it raises, of course, very pointedly this question: Whatever the mathematical interest and the academic value of Mr. Bruhn's paper may be—and these are undoubtedly very great—what is its practical value to shipbuilders, in enabling them to determine, with greater accuracy than is now possible, the strength requisite for the transverse parts of the structure of a ship? I do not myself think, although this theoretical investigation is extremely interesting, and very suggestive in its bearing upon general questions that relate to the strength of ships, that the figures arrived at by it could be adopted in practice. The results of calculations that are limited to statical or still-water conditions, where the forces are fixed both in amount and direction, are entirely over-ridden by the changes in the magnitude and the direction of the forces that come into play when a ship is at sea and rolling among waves. In that case, there are not only the alterations of the statical forces, and the further strains put upon the structure by the shocks of waves, which are referred to in the paper—the amounts of which we have no means of calculating—but there is also the additional and very considerable straining action set up transversely by the inertia of the various parts of the structure and of the loads upon them, which tends to cause distortion of the section formed by the floors, frames, beams, and decks as the ship rolls from side to side. It is the experience arrived at, as the result of watching the effect of the complicated straining actions referred to, upon ships' structures during the whole of their working existence, that has determined the scantlings now adopted for the transverse framing; and, although I do not desire to say anything to depreciate the value of Mr. Bruhn's work (for I put a very high value indeed upon it), I must still express the opinion that Mr. Bruhn and others who may be working in the same direction, have a great deal of hard and difficult work left to do before they will arrive anywhere near a definite solution of this difficult and complicated problem. I would like to add my own expression of thanks to that of Mr. Martell to Mr. Bruhn for being, as I have said, so bold as to grapple with this very difficult subject, and for bringing the results of his investigations before us in such a clear and interesting manner.

Mr. J. Foster King (Member): My Lord, Ladies, and Gentlemen, it is with considerable diffidence that I venture to speak upon Mr. Bruhn's paper, in view of the short time at our disposal for examining such a complicated subject; but there are, I think, one or two points which might tend perhaps to a more perfect understanding of Mr. Bruhn's results, were he to be good enough to

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offer some explanation with regard to them. There is one point in particular on page 284, where it is stated that, "Although the pressure of the water is somewhat in excess of the weight of structure and cargo, the floors might, therefore, be expected to bend inwards; the pressure on the sides is sufficient to completely reverse this bending tendency, so that the largest bending moment on the girder is at the centre of the floors and is tending to bend the floors outwards." Then, again, on page 287: "The frames may also be supposed to be held rigidly fixed at the lowest complete tier of beams, in particular when the beams are loaded with cargo, and are therefore tending to bend the frames in a direction contrary to that due to the pressure of the water." I think it would be of advantage to us, gentlemen, if Mr. Bruhn were, in his reply, to make clear the reasoning whereby he finds that a frame (or floor) subjected to water pressure from the outside, forcing it inwards, having its work as a stanchion superimposed thereon (work which also takes effect in bending) should therefore tend to bend in the direction the reverse to that in which the major force has already directed it, and towards the strong side of the stanchion. To my mind, Mr. Bruhn's conclusions would seem to be the reverse of what one would expect from both practical and theoretical considerations. There is also a reference on page 287 which, so far as I understand it, seems to say that the plating has no effect upon the structural value of the framing. Perhaps I misunderstood Mr. Bruhn, but, if that is his meaning, it seems to me to be impossible to disassociate the plating riveted to the transverse material from the strength of that material; that is to say, the plating must in all cases act in the same way as a rider on a girder. After all, the principal value to shipbuilders and naval architects of such papers as this lies in the practical effect they may have upon the structure of future ships. In the more simple parts of the paper, Mr. Bruhn has come to certain practical conclusions, with which I think most naval architects will agree, and which are worth emphasising, namely, that hold stringers supported by efficient wide-spaced beams are as much entitled as complete decks to be regarded, from a structural point of view, as fixed points; that web frames take up the whole transverse work upon a ship's side, the intermediate frames being only called upon to stiffen the plating between the side girders, and the side stringers transmitting the work from the intermediate frames to the web frames. The value of side stringers in holds, as affecting transverse strength in vessels not having web frames, is measured by their capacity to resist deflection between bulkhead and bulkhead; or, in other words, the ordinary side stringers fitted in merchant steamers are useless factors in transverse strength from any other point of view than that of tying the plating to the framing. Mr. Bruhn deals with the great assistance to the structure afforded by the transverse bulkheads; but, if I may be permitted to say so, argues to a wrong conclusion from unsound premises in saying that, because sailing ships have no bulkheads, therefore racking strains are relatively small. He also says that the framing of sailing ships is relatively larger than that of steamers. I presume that is from the standpoint of the ordinary practice of his society; but that, I think, is scarcely correct, because in ordinary modern sailing vessels, so far as I remember, the frames are actually considerably smaller than those in steamers of the same depth—that is, in two-deck steamers of from, say, 24 ft. to 27 ft. depth. If that is so, then the argument should rather be that steamers' frames should be reduced in size. I am loth to weary you, gentlemen, but this subject is one in which I have taken a very deep personal and professional interest, from the fact that it was my fortune some ten years ago to assist the late Professor Jenkins in carrying out investigations on this very subject, on the lines followed by Mr. Bruhn, but on the lines indicated in the last paragraph of his paper, where account was taken of all the conditions that could be thought of. I have vivid recollections of the enormous amount of work involved, and, like Professor Elgar, I am afraid that we were forced to the conclusion that

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mathematical investigations carried out on lines where so many assumptions have to be made, each of which is open to argument, where so many unknowns must always remain unknown, and where the conditions are so incapable of absolute definition, could not be successfully finished; in short, it is almost hopeless to expect from the calculus practical sizes for the transverse framing of ships.

Professor J. H. Biles (Member of Council): My Lord, Ladies, and Gentlemen, I only want to say one word, and that is of commendation to Mr. Bruhn, an old student of mine, on the admirable work that he has given us in this paper. Of course, work of this kind must necessarily be academic. The consideration of the stresses that come upon a solid of three dimensions, by considering them in three planes at right angles to each other, taking one plane at a time, must necessarily be an academic study. But this is, unfortunately, the only way in which we can attack the problem, unless we are prepared to build ships of fair size, and experiment upon them by taking them out of the water and submitting them to forces that are analogous to those that come upon them at sea. Therefore, while the deductions from this paper must necessarily be carefully and cautiously made, the investigation is an elegant one, and is one that will help students in the future to push further out into the unknown regions of the strength of ships' structures. I am personally very gratified that Mr. Bruhn has taken the trouble he has to put this elegant investigation with its results, and the work it involves, before us.

Mr. K. Meldahl (Visitor): My Lord, Ladies, and Gentlemen, the remarks I shall make will be very short. On page 274 the bending moment and the resistance to bending as affecting these calculations are taken in the usual way, considering the girder as being straight to start with. Now, if you come to look at the bilge of the ship, you have a girder that is being bent and has a crooked shape to begin with. As we all know, the neutral line in that case does not go through the centre of gravity of that section, but, as far as I remember—I have only looked at it for five minutes this morning—it goes somewhat nearer to the outside of the ship, and more so the greater the curvature is in proportion to the depth of the girder. Now, if you take a curvature long and slender in comparison to the depth of the girder, such as in a boat's davits, then you can practically neglect the difference between the two neutral lines; but, if you take a cargo trader of the usual type, with a fairly full 'midship section, I think you will find that the depth of the girder here, as demanded by the rules and practice, will be rather considerable, and, for this part at all events, it seems to me that the calculation would be affected somewhat by this difference. I have no doubt Mr. Bruhn will be able to say whether there will be a great difference caused by this point, because it might affect the results throughout the curves derived from this calculation.

Mr. J. Bruhn, B.Sc. (Member): My Lord, Ladies, and Gentlemen, I think there is a slight misunderstanding with regard to what I had been attempting in this paper. I do not contend that the proposed method is an absolutely mathematically correct solution of the problem. That, I quite agree with Dr. Elgar, is impossible; but, on the other hand, ships have to be built, and the scantlings have to be based on the dimensions of the ship (including draught, length, and depth of holds, &c.), and there must be some relation between the dimensions and the scantlings, I think, whereby we should be able to estimate the necessary strength. What I have attempted in this paper is to use the principle of least work to determine certain forces which would have to be known in even an estimate of the relative strength of the structure; and, in this respect, I think the method would be fairly safe to employ. With regard to the other assumptions mentioned by Dr. Elgar as

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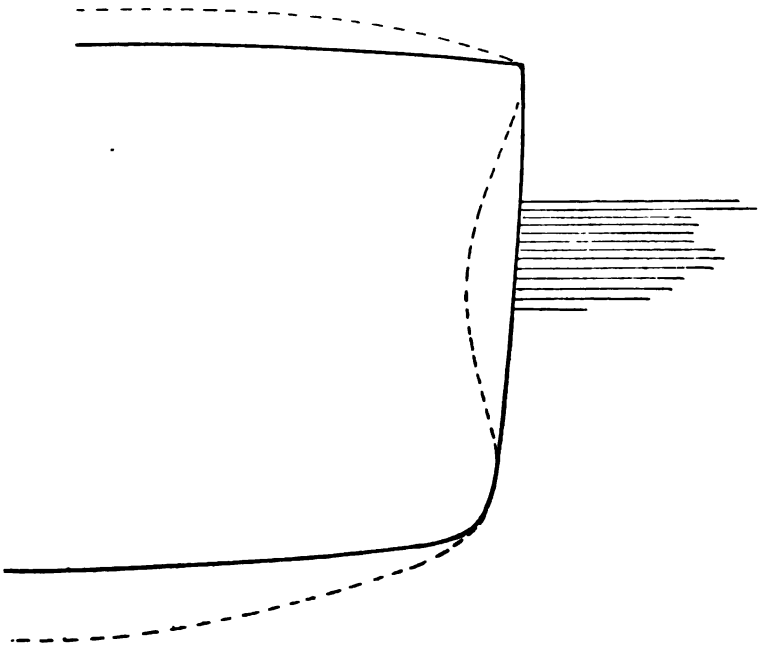
being necessary, I do not see that any other assumptions would be required, beyond those made as to the support lent to the frames by the stringers and plating. If these parts of the structure did not affect the frames, then the whole problem would, I think, be solved by the method of least work. The fact that we can, under those circumstances, arrive at definite values for the forces, seems to me to show that no other assumptions are necessary. The calculations have been made for still-water conditions only, but the method would be equally applicable to other conditions. It is only a question of knowing the forces of the waves striking the ship, and the inertia and reactions of the cargo and structure; they can then all be included in the proposed method. Mr. King spoke of the direction of the bending of the frames. This can be illustrated by the following sketch. If no forces acted on the section, except the pressure of the water on the side of the vessel, then it would tend to assume the form of the dotted line in the sketch, *i.e.*, the frames would be bent inwards at the side of the vessel, and the floors and beams outwards. As the cargo resting on the floors practically neutralises the pressure of the water on the bottom, this is actually what takes place, and I think it is what might be expected. The effect of the plating on the transverse strength cannot, I think, be very great. The plating is very thin and bends very easily, and I do not believe any great stresses can, through it, be thrown on the framing, or that it can support the latter to any appreciable extent.



Mr. Archibald Denny: When you calculate the moment of inertia of frames, that does not include the plating?

Mr. Bruhn: Yes, the plating is included.

Mr. King: I understood you did not include the plating.

Mr. Bruhn: Certainly, it is included. With regard to sailing vessels, I did not say that there were in their case no racking forces because they had no bulkheads. What I did say was, that they proved that their framing was sufficient without any bulkheads, and that the racking stresses cannot, therefore, be very great, because the stresses are great, even without any due to racking. As to Mr. Meldahl's remarks about the position of the neutral axis, I do not think that the effect due to the curvature at the bilge would be great. Some slight modification would, no doubt, have to be made in the calculations to be strictly correct, but I do not think it would affect the results to any great extent.

The President: Ladies and Gentlemen, I beg that you will join with me in giving a hearty vote of thanks to Mr. Bruhn for his highly interesting paper.



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# A SOLUTION OF THE VIBRATION PROBLEM.

By J. H. Macalpine, Esq., Member.

[Read at the Summer Meetings of the Forty-Second Session of the Institution of Naval Architects, June 26, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

## 1. Introduction.

At the recent meeting of this Institution in London, in the discussion of Professor Dalby's able paper, I put before you a special form of balanced engine. I now have the honour of submitting to you fully my reasons for claiming that it is a complete and practical solution of the balancing problem for steamships.

My attention was first seriously directed to the vibrations of steamships by Mr. Yarrow's admirable paper read before this Institution in 1892. It appeared to me, however, that only a portion of the problem was there considered. A few years previously I had made the harmonic analysis of the inertia of a mass moving with the crosshead of an ordinary connecting-rod engine. For a connecting rod four cranks in length this is—

$$F = \frac{M\omega^2 R}{g}(\cos\theta + \cdot 2540\cos 2\theta - \cdot 0041\cos 4\theta + \cdot 00007\cos 6\theta + \&c.)^* \qquad (1)$$

The cranks are supposed to turn uniformly.

$F$ = the inertia in pounds' weight.
$M$ = the moving mass in pounds.
$\omega$ = the angular velocity of the crank in radians per second.
$R$ = the crank radius in feet.
$g$ = the intensity of gravity = 32·2.
$\theta$ = the angle by which the crank has passed top centre, the engine being vertical inverted, as I shall suppose all through this paper.

The second factor of equation (1) is an infinite series, but the only term considered by Mr. Yarrow is the first, viz., $\cos\theta$. On remarking to Mr. Yarrow that the complete series should give rise to other and shorter period vibrations than those occurring at the same speed as the engine, he at once said that this explained the peculiar form of wave produced by the vibrations of the boat; these waves having sharp crests and well-rounded hollows.

Thus, from the first, we have strong evidence of the marked presence of those

* See *Engineering*, Vol. LXIV. p. 512, equation (10); or, Journal of the American Society of Naval Engineers, Vol. IX. p. 508, equation (78).

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short period vibrations arising from the engine—for in many of Mr. Yarrow's experiments the propeller was disconnected.

In the same year, 1892, Mr. H. C. Flood and I made observations on the *Circassia*,* and found, not only vibrations of the same period as that of the revolution of the engine, but also, very prominently, those of one-half and one-fourth of that period. These we called the 1st, 2nd, and 4th period vibrations. There were also marked indications of vibrations of the 6th and 8th period, but no trace of any odd period. This is precisely what we would expect from equation (1), which contains, besides cos $\theta$, only the cosines of even multiples of $\theta$. In several other respects, detailed in our paper, the observations confirmed what theory would lead us to expect; and I think they leave no doubt that these vibrations originate in the engine.

Scarcely has there been a vibration diagram published which has not borne the clearest evidence of those of short period; and no one who has followed the development of this whole subject could fail to be struck by the growing recognition of their importance. Indeed, in two cases which came under my own observation, and in one which was observed by a friend, the 2nd period vibration was much more prominent than that of the 1st period.

I maintain that the short period forces *must* be balanced in any completely satisfactory solution of the vibration problem; and this *cannot* be done with the ordinary type of direct connected marine engine.

The four-crank engine with the two forward cranks opposite, and also the after cranks opposite but at right angles to the former, was introduced for better balance. But, so far as 2nd period forces and couples are concerned, it is exactly equivalent to an engine in which the two forward cranks pass top centre and the two after cranks pass bottom centre at the same time—thus producing the largest possible 2nd period couples. It produces alternately the maximum free force and unbalanced couple for the succeeding higher periods.

A four-crank engine exactly balanced for 1st period is not greatly better than this in regard to 2nd period couples, and has various degrees of unbalance for succeeding periods.

That unbalanced couples are a potent cause of vibration is amply proved by the experiments on H.M.S. *Powerful* and *Terrible*,† and by other experience.

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* See *Engineering*, Vol. LVIII. p. 53.

† "Recent Trials of the Machinery of Warships," by Sir Albert John Durston, K.C.B., R.N., M.Inst.C.E., and Henry John Oram, R.N., M.Inst.C.E.; Minutes of Proceedings of the Institution of Civil Engineers, Vol. CXXXVII. p. 21[illegible]

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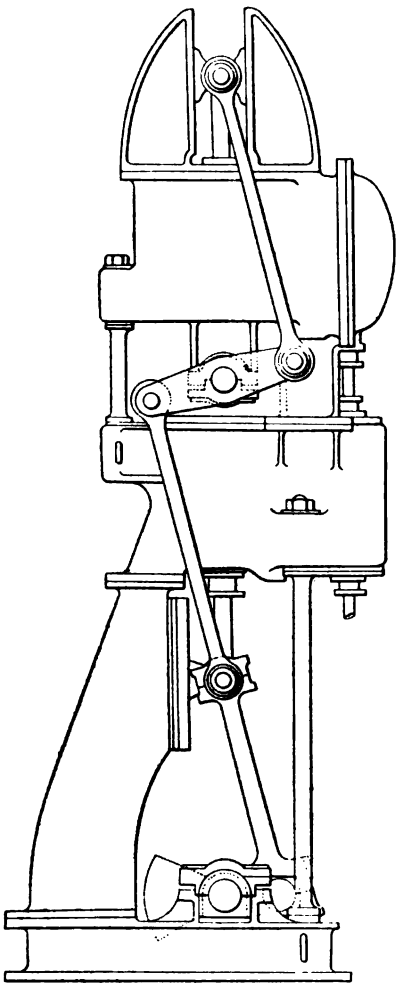
With five and six crank engines we can balance for 1st, 2nd, and 4th period forces, the arrangement being the worst possible for the 6th period. But the engine is complex and costly, and takes up much valuable space in the ship.

Larger numbers of cranks are, I think, absolutely inadmissible.

FIG 1



In all these engines the partial balance is produced by the interaction of the forces from the various cranks, the ship being considered rigid under the engines, though the evil it is sought to cure arises from the flexibility of the ship. The assumption of rigidity is only approximate, and thus the presence of very sensible 1st period vibrations, in diagrams taken from ships whose engines are balanced for this period, may very probably be accounted for.

2. ONLY ONE CONDITION FOR COMPLETE BALANCE.

There is one way—and only one—in which an exact balance for the vertical inertia forces can be attained. That is, by adding to the right-hand side of equation (1) an expression—

$$-\frac{m\,\omega^2 r}{g}(\cos\theta + \cdot 2540\cos 2\,\theta - \cdot 0041\cos 4\,\theta + \cdot 00007\cos 6\,\theta + \&c.) \quad (2)$$

If then—

$$\mathrm{M\,R} = m\,r, \quad (3)$$

we get—

$$\mathrm{F} = 0. \quad (4)$$

3. FORMS OF ENGINE IN WHICH THIS SOLUTION IS REALISED.

This solution is realised, with the most complete success, in small fan engines, where two cylinders are placed exactly in line, but on opposite sides of the crank shaft. One piston actuates a central crank, and the other piston two cranks—one on each side of, and at 180° to, the first. But the arrangement is entirely unsuitable for shipboard.

This exact balance can, however, also be obtained if we place the cylinders in line on the same side of the crank shaft; one piston being directly connected to the crank, and the other connected through a lever to the same crank. This is roughly shown in the accompanying illustration (Fig. 1). If the inertia of the crank and a portion of the connecting rod is balanced by a light counterbalance, as shown, and if the moving masses are properly adjusted, both horizontal and vertical balance is exact.

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Figs. 2 to 13, Plate LIV., give a design in which this exact vertical and horizontal balance is attained in a manner much more applicable to marine engines. The engine is more unbalanced torsionally than the ordinary type of inverted marine engine. But I will show in §§ 9 to 11 :—

(*a*) That the period of slowest torsional vibration of the ship is so short, compared with the engine revolution, that the resulting vibration is a rigid body rotation of 1st period, almost entirely, and of little more than infinitesimal amplitude.

(*b*) That the excess of torsional couple is confined, almost exclusively, to the 1st period. Those of high period which, by synchronism, might set up elastic torsional vibrations, being equally feeble in the ordinary and proposed designs. Thus, as elastic torsional vibrations are not set up by the former, they will not be set up by the latter engine.

I will now give a few figures and explanations referring to the design, Figs. 2 to 13, Plate LIV. These, with a few additions, are the same as were submitted at the end of last year, along with the design, to my friend Admiral George W. Melville, Engineer-in-Chief of the United States Navy. I am very proud to be able to say that this distinguished engineer, after careful examination, officially signed the design, and writes: "I have full faith in it, and have approved it"; and, further, declared his intention, should opportunity offer, of adopting it.

Cylinders, 27 in., 39 in., 56 in., and 80 in. diameter.
Stroke, 48 in.
Boiler pressure, 200 lbs. per square inch.
Designed piston speed, 900 ft. per minute.
Revolutions, 112·5 per minute.

### 4. Calculations for Balance.

(*a*) Connecting rod* :—

| | |
|---|---|
| One connecting rod ... ... ... ... ... ... ... ... | = 3,648 lbs. |
| Length between centres, 9 ft. ... ... ... ... ... ... | = 108 in. |
| Centre of gravity of rod from crosshead centre ... ... ... ... | = 76·38 in. |
| Part of connecting rod to be taken as revolving with crank pin ... ... ... ... ... ... ... | $= 3{,}648 \times \frac{76{\cdot}38}{108} = 2{,}580$ lbs. |
| Part of connecting rod to be taken as reciprocating with crosshead ... ... ... ... ... ... ... | $3{,}648 - 2{,}580 = 1{,}068$ lbs. |

* For the reason of this calculation, see "Analysis of the Inertia Forces of the Moving Parts of an Engine," by the present writer; *Engineering*, Vol. LXIV. p. 513, remarks after equations (15) and (17); or, Journal of the American Society of Naval Engineers, Vol. IX. p. 513, remarks after equations (83) and (85).

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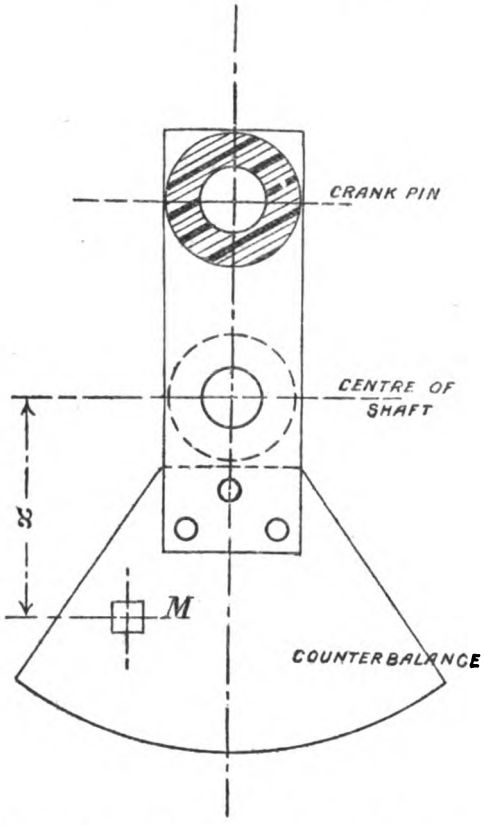
(*b*) Counterbalancing the parts revolving with the crank :—




| | Mass. Lbs. | Arm for Leverage. Inches. | Moment. Inch-pounds. |
|---|---|---|---|
| Part of connecting rod to be taken as revolving with crank pin—from (*a*) ... ... ... | 2,580 × | 24 | = 61,920 |
| Crank pin, taken as solid ... | 1,158 × | 24 | = 27,792 |
| Crank webs on crank pin side of shaft centre... ... ... | 2,930 × | 14·1 | = 41,313 |
| | | | 131,025 |
| Deduct for hole at crank radius | 579 × | 24 | = 13,896 |
| Total moment ... ... ... | | | 117,129 |
| Two counterbalances and parts of crank webs on same side of shaft centre* ... ... | 5,250 × | 22·3 | = 117,075 |

The counterbalances thus exactly balance the other parts revolving with the crank, as the inertia forces are equal and oppositely directed in the same plane.

(*c*) Reduced mass to be taken for levers.

As all parts of the main levers have not the full 48-in. travel, they must be taken at a reduced weight, the moment of each end being reckoned from the rocking centre, and the arm being measured parallel to the centre line of the lever, similarly to the calculation for the counterbalances in (*b*), above.

| | Crank End of Lever. | Outer End of Lever. |
|---|---|---|
| Length between rocking centre and end centres ... | 39 in. | 39 in. |
| Mass ... ... ... ... ... ... ... | 2,482 lbs. | 2,123 lbs. |
| Moment ... ... ... ... ... ... ... | 54,604 in.-lbs. | 44,160 in.-lbs. |
| Arm = moment ÷ mass ... ... ... ... | 22 in. | 20·8 in. |
| Reduced mass ... ... ... ... ... ... | $\frac{2,482 \times 22}{39}$ = 1,400 lbs. | $\frac{2,123 \times 20·8}{39}$ = 1,132 lbs. |

(*d*) The parts moving with the crosshead of the high pressure cylinder must now balance those of the 2nd intermediate pressure; similarly, the corresponding parts of

* Obviously, in counting the moment for the counterbalances, the sum to be taken is $\Sigma M x$, where M is any small portion of the mass and $x$ its distance from the shaft centre, measured parallel to the crank radius. See above sketch.

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the 1st intermediate pressure and low pressure must balance, as in the following table :—

| —— | High Pressure. | 2nd Intermediate Pressure. | 1st Intermediate Pressure. | Low Pressure. |
|---|---|---|---|---|
| | lbs. | lbs. | lbs. | lbs. |
| Piston ... ... ... ... ... | 662 | 2,329 | 1,416 | 3,083 |
| Piston rod ... ... ... ... ... | 1,180 | 1,428 | 1,180 | 1,428 |
| Guide block ... ... ... ... | 1,425 | 689 | 1,425 | 689 |
| Part of connecting rod. See (*a*) ... | 1,068 | — | 1,068 | — |
| Links to levers ... ... ... ... | 1,067 | 1,224 | 1,067 | 1,224 |
| Reduced mass of levers ... ... ... | 1,400 | 1,132 | 1,400 | 1,132 |
| Total ... ... ... ... ... | 6,802 | 6,802 | 7,556 | 7,556 |

The vertical forces from these parts are now exactly balanced, and are zero. The piston rods and crossheads of the high pressure and 1st intermediate pressure are shown bored out, merely to call attention to the fact that the balance may be obtained by *deducting* weight as well as by the less costly way of adding weight, say to the pistons, where necessary.

In the design shown, it has not been necessary to depart from the usual proportions of the parts to effect a balance.

(*e*) The parts of the main levers, and links at their ends, have a small transverse movement, principally of 2nd period; but the forces from this are also balanced thus :—

| | Crank End. | Outer End. |
|---|---|---|
| Mass of links ... ... ... ... ... | 1,067 lbs. | 1,224 lbs. |
| Length of links ... ... ... ... ... | 55 in. | 55 in. |
| Distance of centre of gravity of links from upper ends ... ... ... ... ... | 27·5 in. | 35 in. |
| Reduced mass of links for estimating horizontal forces ... ... ... ... ... ... | $\frac{1,067 \times 27·5}{55}$ = 533·5 lbs. | $\frac{1,224 \times 35}{55}$ = 779 lbs. |
| Reduced mass of levers from (*c*) ... ... | = 1,400 lbs. | = 1,132 lbs. |
| Total ... ... ... ... ... | 1,933·5 lbs. | 1,911 lbs. |

To produce this balance, the extremity of the outer end of the lever has been thickened to 5½ in., whereas, 4 in. is ample for strength. By a very small further addition, the totals 1,933·5 and 1,911 could be made to agree exactly; but they are left, in order to indicate that an infinitesimal residuum need not be taken account of.

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The accuracy of this balance can readily be checked after the parts are made. By suspending the links, the positions of their centre of gravity may be found, and the reduced mass calculated exactly. Suspending weights equal to these respective reduced masses, at the outer end and crank end centres, should make the levers balance on their central axis. The additional weight required to do this will give at once the defect of balance.

(*f*) Mode of procedure.

If the following sequence be observed in making drawings and calculations, no further delay need be incurred in the design than the time required to make a few simple calculations.

(1) After the crank shaft and connecting rod are drawn, calculate and draw the counterbalance of the requisite size.

(2) After the main levers and their links are drawn, make any slight addition required to one or other end of the levers to produce horizontal balance, as in (*e*).

(3) After all the other moving parts are drawn, add to one or other of the balancing pistons sufficient to make exact balance, as in (*d*).

This also suggests the proper mode of procedure in building the engine. A little metal should be left on each piston till all the other moving parts are finished and weighed; after which the requisite amounts of metal should be taken off to produce the exact required balance.

(*g*) The balance of the valve gears is obtained in precisely the same way as the foregoing. But it would be ample to make—

| | | |
|---|---|---|
| Reduced mass of ahead eccentric ... | = | Reduced mass of outer end of levers |
| + ahead strap and rod ... ... ... | | + crosshead |
| + (say) ½ link ... ... ... ... | | + valve spindles |
| + ½ drag-rod ... ... ... ... | | + 2nd intermediate pressure valves. |
| + valve spindle ... ... ... ... | | |
| + reduced mass of crank end of levers | | |
| + high pressure valve.... ... ... | | |

And, similarly, for the 1st intermediate pressure and low pressure gear.

The engine is now exactly balanced for all vertical and horizontal forces. Being balanced for all periods, there will be an absence of these short period vibrations, or tremors, that have been found so annoying, and which must make the correct aiming of a gun much more difficult.

## 5. Remarks on Design.

In this engine the balance is complete for each crank by itself; and, therefore, perfect longitudinal flexural rigidity of the ship under the engines is not required for complete success.

I prefer the alternative plans shown in Figs. 5 to 7, Plate LIV. In them the

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cylinder clearances are reduced to a minimum. Working the high pressure and 1st intermediate pressure valves by an overhung arm, as shown, is not novel, and cannot be at all objected to if the vertical guide rods, working in the bearings at the ends of these cylinders, are made stiff enough. This stiffness has been amply provided.

The counterbalance, marked G, shown by dot and dash lines, must be added for an exact balance; but its omission would only leave a very minute couple tending to produce vertical transverse vibrations.

The small plan of the valve gear, Fig. 10, Plate LIV., shows that the valve levers are held from fore and aft movement by the bearings at the rocking centre; and that clearance is provided at the ends of the levers, so that any relative movement due to expansion, by heat, of the two cylinders cannot strain the gear. Transverse expansion is allowed for by a small extra clearance in the slipper blocks of the centre and outer bearings.

Great care should be taken to keep the connections of the high pressure and 1st intermediate pressure cylinders to the framing very strong, as this is not accomplished so readily as for the other two cylinders. These connections, and the ribbing of the cylinder bottoms, are somewhat fully shown. The bolting of the castings, carrying the main guides, to the framing and cylinders should also be very strong, as there is a pull of fully 60 tons on it from the levers.

If the cutting away of the main guides, to clear the levers, is considered too much, or if it is desired to fit dippers, as it generally would be, the levers may be lowered or the connecting rod lengthened a few inches. In the former case, the connection of the counterbalances to the cranks will have to be modified.

The main levers are kept low down to bring them below the casting carrying the main guides. This not only makes them easily portable, but it also gives such length to the lever links that but slight pressure is produced by them on the main guides.

The crosshead bearing of the high pressure and 1st intermediate pressure cylinders is formed in the piston rod. The links from the crank ends of the main levers are thus directly connected to the connecting rod, and the forces from them do not pass through the crosshead bearing. If the crosshead bearings were in the connecting rod, as is frequently the case, these forces would be transmitted through them, thus making a more indirect connection to the crank shafts. The design adopted is also lighter and more compact, and, hence, for all reasons is much to be preferred.

*Valve Setting.*—Although not shown on the drawing, the valve setting preferred is that of the yacht *Sovereign*,* as it gives a much longer range of cut-off than the usual setting.

* See *Engineering*, Vol. LXV. p. 334, and Fig. 5; or, Journal of the American Society of Naval Engineers, Vol. X. p. 611, and Fig. 5.



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Stroke of valves, 8 in., full ahead gear; valve circle B (Fig. 5, *Engineering*), of *Sovereign's* valve setting. When the engine is running at 112·5 revolutions per minute, the speeds of steam in feet per second for this setting are:—

| | H.P. | 1st I.P. | 2nd I.P. | L.P. |
|---|---|---|---|---|
| Inlet ... ... ... | 100 | 156 | 172 | 182 |
| Exhaust ... ... | 52·3 | 81 | 90 | 95 |

Pressures on bearing from steam pressure:—

| | | Lbs. per sq. in. |
|---|---|---|
| Maximum total pressure on each piston ... ... ... | = | 67,400 lbs. = 30 tons. |
| Main bearings ... ... ... ... ... ... ... | = | 200 |
| Crank pin ... ... ... ... ... ... ... ... | = | 440 |
| High pressure and 1st intermediate pressure crosshead bearings... ... ... ... ... ... ... ... | = | 842 |
| High pressure and 1st intermediate pressure guide blocks... | = | 48 |
| Bearings of links to levers ... ... ... ... ... | = | 802 |
| Centre bearings of main levers ... ... ... ... | = | 560 |
| 2nd intermediate pressure and low pressure guide blocks ... | = | 20 |

The combined steam and inertia stress alters the maximum in the bearings by about the same proportion as in the ordinary design.

The pressures are seen to be under ordinary practice; especially in the parts making connection from the cylinders at the back. This involves no clumsiness of design, and, by making the wear of these bearings very slow, will insure smooth working of this gear for long periods without setting up. In the merchant engine design to be referred to immediately, the pressures in the lever connection of the low pressure and 2nd intermediate pressure cylinders, and in the high pressure and 1st intermediate pressure crossheads, are reduced to about 60 per cent. of those just given. This could also be done in the first design, thus removing any possible anticipation of difficulty. The thicker oil film, allowed by this reduced pressure, will reduce the rate of wear very much more than 60 per cent., and will prevent any increase of frictional loss from the increased diameters of the journals. In discussing these designs recently with a friend, who is an engineer of large experience in high speed machinery, he very truly remarked that quiet running is almost entirely a question of large bearing surfaces.

### 6. The Lever Connection.

The point which would claim the attention of any engineer considering the proposed design, is the introduction of the lever. In this country all but the oldest members of our profession have never seen the lever used in the principal parts of marine engines, except in the valve and pump gears. But in America the beam engine is almost universal for large river boats, and often runs at high piston speed. It is well known, and it comes within my own experience, that it is an exceedingly quiet and smooth-running engine, and costs exceptionally little for repairs. Indeed, these are

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the very points which seem to have enabled it to hold its place, in that country, against more compact and lighter direct connected engines.

In Britain, where few but ocean steamers are built, the beam or side lever engine disappeared merely because the screw propeller was adopted. Any remarks I can find with regard to the history of this engine are similar to what I have just said in favour of the American engines.

In the large proportion of machines used in the Arts, whether running at high speeds or not, the lever is introduced with perfect freedom, and with no objectionable result. Though the bearings are most frequently unadjustable, the machines run for long periods without overhaul.

There are two causes from which trouble could arise :—

(1) The increased number of joints between the piston and shaft.

(2) The increased elasticity due to the linkage.

That both sources of trouble *can* be avoided is proved conclusively by the experience, just cited, of beam engines and other machines. But with regard to the number of joints, I will further cite, as one of many instances familiar to marine engineers, the valve gear by Heusinger von Waldegg, also known as the Walschaert, and on the Clyde as the Brock gear. This is an excellent gear, which runs smoothly and quietly in high-speed engines, with much heavier pressures on the bearing surfaces than I have adopted above. The number of bearings is greater, in a considerable proportion, than in the design I propose.

Regarding the effect of elasticity, I have recently communicated to the Journal of the American Society of Naval Engineers * a long investigation on the "Inertia Stress of Elastic Gears." I have shown that where the movement of any mass is frictionally resisted—as, in this case, that of the pistons – we are usually liable, under perfectly definite circumstances, to a large increase of inertia stress through the production of a vibration of natural period—a vibration rarely visible. Thus the fact that the factors of safety in running machinery are much greater than Wöhler found necessary, is completely accounted for, since no such effects of inertia were produced in his experiments. It would take much too long to discuss, even briefly, the principles involved, and I must be satisfied with referring to my previous investigation. But in the design proposed, it can be shown that this increase of stress is scarcely more than for the usual direct connection; and the ordinary calculation of the stresses, with the usual factors of safety, will be sufficient, except for the levers. These are designed for stiffness alone, and are much in excess of the requisite strength. The natural

* Vol. XII.

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period of the low pressure piston, in the design, Figs. 2 to 13, Plate LIV., is about the 30th, when the engine is running at 900 ft. per minute piston speed (= 112·5 revolutions), and this is far above what is absolutely necessary for steady working.

Comparing the indirect connection with the direct one: the main guide and crosshead bearings of the low pressure and second intermediate pressure cylinders are not altered from the usual design, except that the former is greatly, and the latter to a considerable extent, reduced in pressure per square inch from ordinary practice. The outer lever bearing replaces the crank pin of the direct connection, and, similarly, the centre lever bearings replace the main bearings; but they are both much more favourable than the rotating bearings they replace and can be much more closely set up, the diminished slack helping to prevent shock. Against these advantages we have the extra links at the crank ends of the main levers. The character of the reversal of stress in the crank pins and main bearings is in no way altered by having them taking the stress of two cylinders instead of one; and they should work as quietly and well, if, as is the case, the pressure per square inch is kept about the usual amount.

A minor point which might cause anticipation of difficulty is that the lever links stand at an angle to the vertical at the ends of the stroke. Thus, when the steam pressure is reversed at the end of the stroke, there is a comparatively sudden reversal of pressure on the main guide, possibly producing shock; while, in the direct connected engine, the connecting rod being then vertical, there is no reversal of pressure on the guide due to steam pressure. But we must recollect that in high-speed engines the inertia usually rises above the highest stress due to steam, and reversal of pressure on the main guide occurs not only at the end, but again further on in the stroke, when the connecting rod is standing at a considerable angle. As the links from the levers, in the proposed design, are at this second reversal nearly vertical, the conditions are decidedly more favourable than in the ordinary engine at high speed; though, as no trouble arises in it, even at the second reversal, none need be anticipated in the proposed design at high or low speed.

Finally, it may be asked, What is the crank but a lever, and that not of the best kind, with its large span, rotating journals, and bearings whose diameters bear such a considerable proportion to the length of crank? The proposal is to replace these by levers of a better description.

In the early days the crank was long viewed with distrust—which was unfounded; and, I believe, misgivings which some might entertain here will quickly vanish when the proposal has become familiar, and its advantages have been weighed. Indeed, I have already found this to be the case.

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### 7. Advantages of Design.

I claim, then, if there is any sufficient object to be gained, there is no reason why the levers should not be introduced. And the objects to be gained are great: —

We thus realise a complete and unique solution of what may be said to be the great problem before marine engineers for many years: the prevention of vibration arising from the engine. And along with this we have saving of weight, simplification of the engine, and great saving of space.

From the perfect balance we get the secondary advantage of diminished wear and tear in the non-vibrating as against the vibrating machine. Bearings can be adjusted more closely, and the engine can be run with equal safety at a higher number of revolutions.

The saving of weight and simplification of the engine are quite evident from an inspection of the design; but I will mention the leading points :—

(*a*) There are two cranks against four, and these are of practically the same diameter, since the same power has to be transmitted through the after ends, and the curve of turning moments is almost the same as in an ordinary four-crank engine. The extra span of bearings, due to the increase of bearing surface required, might make a very small addition to the diameter desirable.

(*b*) There are two main connecting rods against four.

(*c*) Three main bearings against, usually, eight; as a four-throw crank shaft of any size is most frequently made in four pieces.

(*d*) There is much less framing than in the ordinary design, and that of a very simple form.

(*e*) There is little more than half the valve and reversing gear.

Against these parts taken away, there are levers, with their links and bearings, added; but these are much smaller than the parts they replace.

The saving of space is clearly shown by Fig. 1A, on the following page, where a plan of the cylinders of an ordinary design is shown on one side of the centre bulkhead, and the proposed design on the other side. The short lines *a a* mark the ends of the main bearings. It may seem that, with regard to the breadth of the engine-room, the ordinary design has been unfairly dealt with. But this figure gives a comparison which Admiral Melville wished made with one of the United States ships already in commission. The distance of the centre line of engine from the centre of the ship was 11 ft. as figured, and this distance is no greater here than it is frequently made. No doubt it might be reduced to some extent, still leaving a good platform next the centre bulkhead; and then the ordinary design would have a little advantage in breadth at the upper casings. It may be noted that in the proposed design the

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framing is formed so as to leave a good passage, under the cylinders, next the bulkhead. At the level of the cylinders, the stuffing box in the steam connection between the 2nd intermediate pressure and low pressure cylinders needs alone to be got at, and it is perfectly accessible for packing and adjustment.

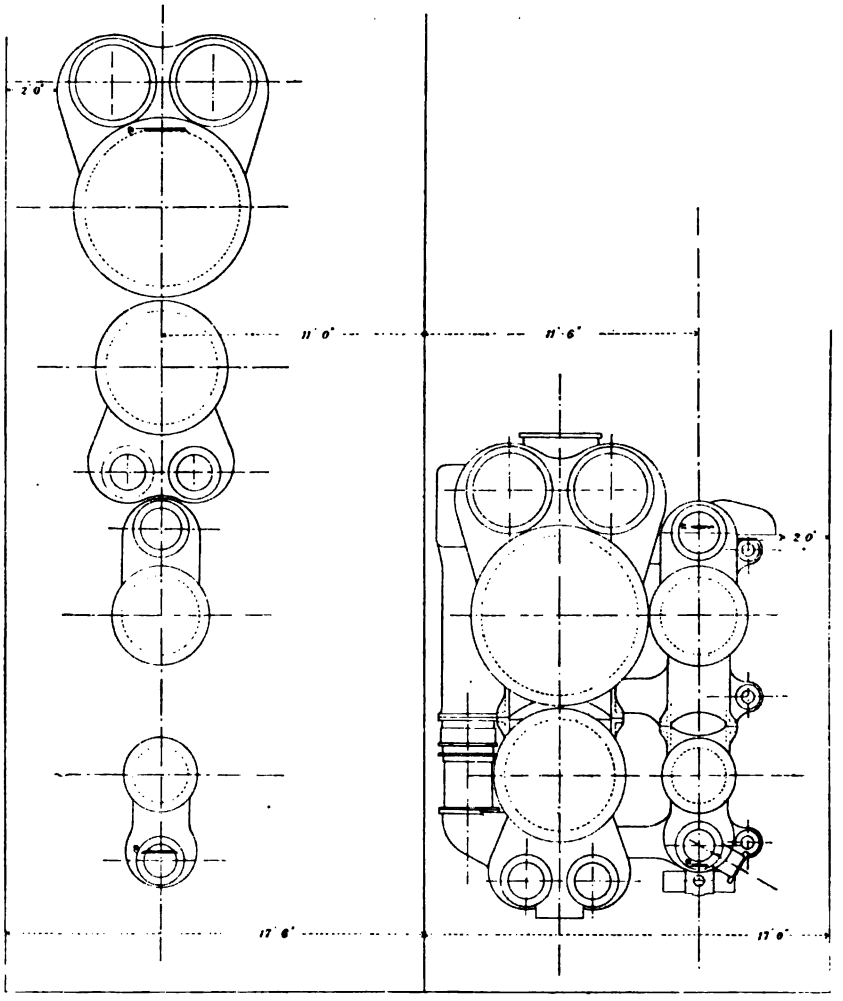


FIG. 1A.

The proposed design is little more than half the length of the ordinary, shown in Fig. 1A, even though, in the latter, the crank shaft is supposed only to be in two pieces. Thus the adoption of the former greatly shortens the necessary length of the machinery space. Indeed, Admiral Melville found that in some recently designed ships of the United States Navy, while the four-crank quadruple did not allow of a good arrangement for triple screws, since the machinery space became so long, the very considerable shortening due to the proposed engine would have made the triple-screw design perfectly practicable.

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Although the engine is so compact, all the bearings are accessible when it is running. And, having several of the heavier replaced by lighter parts, it will be the more easily overhauled, its bearings adjusted, and so on. Such questions have been carefully watched in making the design.

## 8. Merchant Engine Design.

Of the merchant engine design shown in Figs. 14 to 19, Plate LV., only a few words need be said.

In merchant ships there is nearly always ample head room over the engines. I have, therefore, placed the back cylinders above the level of the front ones, to allow the levers to be accommodated above the condenser. The front ends of the levers work up at the sides of the front cylinders, and the valve casings are moved up to allow of this. The valve movement is, however, supposed to be the same as for the previous design.

The non-rotary bearings in this design have double the surface usually given to bearings of this character, such as crossheads: the pressure from steam on the centre lever bearings is only 350 lbs. per square inch. It will, therefore, be possible to make the lubrication very perfect and the wear exceedingly slow.

Ample rigidity is retained in the lever connection to the back cylinders.

Strong frames are cast on the top of the condenser and the strength carefully carried down to the bottom. On these frames the centre bearings of the levers, and the cast brackets which take hold of the backs of the front, and the bottoms of the back cylinders are supported. These brackets are L shaped in horizontal section, in order to afford ample athwartship and longitudinal stability to the engine. Between the 2nd intermediate pressure and low pressure cylinders, at the back, instead of a bracket, there is a wrought iron column to the condenser, and a strong triangular frame connects the front and back cylinders. Thus the centre lever bearings, between the two pairs of levers, are left quite accessible.

The bolts holding the caps of the centre lever bearings are intended to pass down to, and pull direct on, the condenser casting. By dropping these bolts down, lifting the levers, removing the plumber blocks, the back guide bar, and also the piston rod of the front cylinder, the levers may be drawn out towards the front. This is more work than in the previous design, but it is an operation of no great difficulty, and may not be required once in the life of the engine.

Marshall valve gear is shown, though the Stephenson link motion would be equally convenient. The counterbalance marked G on the previous plan, Fig. 5, Plate LIV., is not shown here; though, of course, it is equally necessary for an exact balance.

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The air, feed, circulating, and bilge pumps are worked by levers on the extended rock shafts of the main levers. Any of these pumps could be fitted in duplicate, if desired. The moving parts are here, also, balanced for vertical and horizontal forces.

The angular motion of the pump links is so small, that I have made the guides for the rods and plungers in the pumps themselves; partly by a long neck bush, and partly by the gland. This latter has an upper part of larger diameter, which fits the pump body accurately, and is entered a short distance before the lower and smaller part, which performs the usual function of retaining the packing, begins to enter the stuffing box. The guide is thus made comparatively long. I have used this successfully in similar cases where the guidance required was not very great. If preferred, ordinary guides may be fitted, but they will require to be removed to withdraw some of the condenser tubes. As the pumps stand, any tube can be withdrawn whenever the condenser doors are removed.

Possibly it would have been better to extend the engine a few inches, as the condenser casting has to be slightly cut away to clear the levers at bottom stroke; but not to such an extent as to affect the strength.

Figs. 14 to 16, Plate LV., show the steam connections, which, for clearness, are omitted from the more detailed plans. They are arranged to diminish the breadth of engine as much as possible. With the low pressure exhaust as shown, the 1st intermediate pressure valve would have to be withdrawn downwards; but there is ample room to do this.

The engine is not only much shorter, but is also narrower below than the ordinary merchant design with the pumps at the back of the condenser.

In the majority of cases, especially for high-class merchant work, I would prefer the first design.

I will now treat the questions relating to the unbalanced torsional couples from the engine and the torsional vibrations of the ship; which, being more mathematical, I have intentionally reserved to the end of the paper, merely stating the general results earlier.

### 9. Torsional Vibrations of the Ship.

The ship's skin, decks, a proportion of the frames and stiffeners, &c., contribute to the torsional rigidity of the ship. The woodwork, machinery, and cargo, while increasing the moment of inertia, at each section, about the centre of vibration, do not sensibly add to the elastic re-action. As the skin and decks will mostly lie at a relatively great distance from the centre of rotation, the proportion they contribute to

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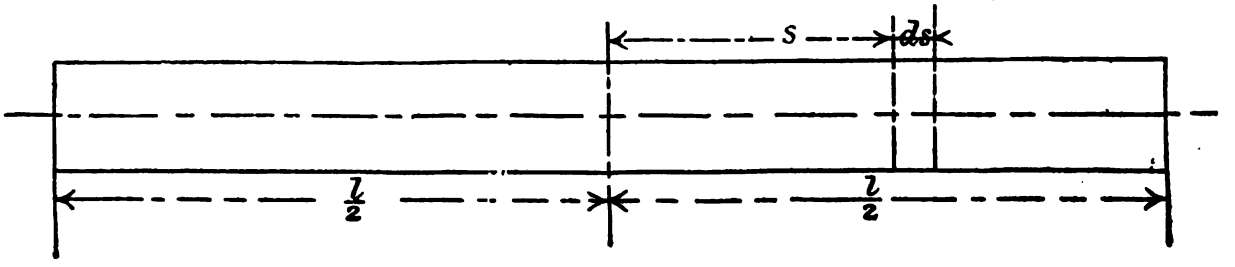
the moment of inertia is evidently much larger than the proportion they form of the total weight of the ship; especially as in calculating moment of inertia we have to deal with the square of the radius. Loose stores and cargo would not participate fully in torsional vibrations, and hence their weight should be somewhat discounted. Indeed, by slight relative motion of their parts, they will produce friction, and thus absorb some of the energy of the vibrations; thereby contributing to damping them out. Woodwork, and even the viscosity of the steel structure, will have the same effect. To see how small this damping effect needs to be to absorb very short period vibrations, I refer to Chapter XIII. of the "Inertia Stress of Elastic Gears," previously mentioned.

From figures given me by my friend Professor F. P. Purvis, I conclude that the proportion, by weight, of a ship loaded, which contributes to torsional rigidity varies from a lower limit of about one-eighth, for merchant vessels, to one-fourth or one-fifth for war vessels. When, in merchant vessels, the proportion is so low as one-eighth, about 70 per cent. of the total loaded weight will be cargo, contributing, as noted above, but slightly to the moment of inertia in proportion to its weight, since its radius of gyration is small relatively to the decks and skin, and it will not fully participate in the torsional vibration. Hence, I think, we may safely conclude that the total moment of inertia of a ship will range from a higher limit of about three times that due to the parts giving torsional rigidity, to less than twice; the figures for warships being nearer the lower limit. In what follows, the periods of torsional vibrations are given for an increase of moment of inertia of two, three, and four times, and it will be seen that even the latter high value in no way invalidates my argument.

To take the very simplest case, we may calculate the speed of torsional vibration of a thin steel tube. Suppose torsional couples applied to the centre of this tube, as the engine placed amidships will apply couples to the ship. The slowest elastic torsional vibrations will be those with two nodes, each one quarter of the length of the tube from the ends.

$I$ = moment of inertia of section of tube, augmented as above, per foot length.
$s$ = distance of any transverse section from the centre of length of the tube.
$t$ = the time in seconds.
$\theta$ = the angle, in circular measure, by which this section is turned from its position of equilibrium, at time $t$.
$M$ = the moment of torsional elastic forces about the longitudinal centre line at section $s$.

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The pound is the unit of mass, the pound weight the unit of force, the foot the unit of length, and the second the unit of time.

The moment of inertia of a portion between sections $s$ and $s + ds$ is $I\,ds$. The acceleration of the rotation of this section is due to the difference of the moments at its two ends. Hence we have—

$$\frac{I\,ds}{g}\frac{d^2\theta}{dt^2} = dM. \tag{5}$$

Also—

$$M = C\frac{d\theta}{ds}, \tag{6}$$

where C is the couple which would give the tube unit rate of twist (supposing the elastic limit not passed).

From (6)

$$dM = C\frac{d^2\theta}{ds^2}ds. \tag{7}$$

From equations (5) and (7)

$$\left.\begin{aligned} \frac{I}{g}\frac{d^2\theta}{dt^2} &= C\frac{d^2\theta}{ds^2}, \\ \text{or} \qquad A^2\frac{d^2\theta}{dt^2} &= \frac{d^2\theta}{ds^2}, \end{aligned}\right\} \tag{8*}$$

where

$$A^2 = \frac{I}{Cg}.$$

The solution of equation (8), which, we will see, suits all conditions, is—

$$\theta = K\sin\frac{2\pi}{Al}t\cos\frac{2\pi}{l}s, \tag{9}$$

where K is a constant.

Conditions :—

(1) $\theta = 0$ for $t = 0$ at all parts of the bar; that is, $t = 0$ is chosen for the time when the bar is passing through its configuration of equilibrium.

(2) $$\frac{d\theta}{ds} = -K \times \frac{2\pi}{l}\sin\frac{2\pi}{Al}t\sin\frac{2\pi}{l}s = 0 \text{ for } s = 0.$$

This condition is obviously required by symmetry, as the vibration must be a maximum at the centre of the tube.

(3) $$\frac{d\theta}{ds} = 0 \text{ for } s = \pm\frac{l}{2};$$

* For equation (8) see also Love's "Theory of Elasticity," Vol. II. p. 109, equation (17).

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that is, at the ends of the bar. This must hold, since there is no mass beyond to produce any elastic moment by its inertia. Hence the rate of twist must be zero.

K is undetermined, as the vibration may be large or small.

As stated above, there is a node at quarter of the length of the bar from each end, for equation (9) gives—

$$\theta = 0 \text{ for } s = \pm \frac{l}{4}.$$

In the time T of a complete vibration, $\frac{2\pi}{Al} t$ must increase by $2\pi$. Hence

$$\frac{2\pi}{Al} T = 2\pi, \text{ or } T = Al. \qquad (10)$$

The moment of inertia of the tube unloaded is—

$$2\pi r \tau \times r^2 \times \rho = 2\pi\rho r^3 \tau,$$

where—

$r$ = the radius.
$\tau$ = the very small thickness.
$\rho$ = the density of steel.

If this moment of inertia is augmented $n$ times, as explained above, we have—

$$I = 2\pi n \rho r^3 \tau. \qquad (11)$$

To estimate C :—

If the radius was unity, unit rate of twist would change a line on the surface of the tube, which originally was straight and parallel to the axis, into a spiral of angle $\phi = 45°$, or $\tan\phi = 1$; and the shearing stress per unit section would be $\mu$, the modulus of rigidity. For radius $r$, unit rate of twist would thus evidently give—

$$\tan\phi = r, \text{ and shearing stress} = \mu r.$$

Hence, the moment C is due to stress $\mu r$ over section $2\pi r \tau$ at radius $r$. Thus we get—

$$C = 2\pi r \tau \times r \times \mu r = 2\pi\mu r^3 \tau. \qquad (12)$$

$$A^2 = \frac{I}{Cg} = \frac{2\pi n \rho r^3 \tau}{2\pi g \mu r^3 \tau} = \frac{n\rho}{g\mu},$$

from equations (11) and (12).

Therefore, from equation (10)

$$T = l\sqrt{\frac{n\rho}{g\mu}}. \qquad (13)$$

The speed of vibration is seen to be independent of the diameter of the tube.

$\rho$ for steel = 500 lbs. per cubic foot.
$\mu$ " " = $12 \times 10^6$ lbs. per square inch.
= $12 \times 144 \times 10^6$ lbs. per square foot.
$g$ = 32·2.

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For a tube (ship) 400 ft. long we get from equation (13),

$$T = 400\sqrt{n}\sqrt{\frac{500}{32{\cdot}2 \times 12 \times 144 \times 10^6}} = {\cdot}03792\sqrt{n}.$$

If N = the number of vibrations per minute—

$$N = \frac{60}{{\cdot}03792\sqrt{n}} = \frac{1582}{\sqrt{n}}. \qquad (14)$$

In a ship 400 ft. long, 112·5 may be taken as near the highest number of revolutions of any engine which will be put in it, even if it is a warship. The number of the synchronising period then is—

$$S = \frac{1{,}582}{112{\cdot}5\sqrt{n}}. \qquad (15)$$

We thus get the following table:—

| | | | |
|---|---|---|---|
| $n$ = multiplier for moment of inertia ... ... ... | 2 | 3 | 4 |
| N = number of vibrations per minute, equation (14) ... | 1,119 | 913 | 791 |
| S = number of synchronising period, equation (15) ... | 9·9 | 8·1 | 7·0 |

For a solid circular bar it can readily be shown that the result is precisely the same. And similar results can readily be worked out for other sections. For instance, for a hollow tube, the sections of which are bounded by two similar, concentric, and similarly placed ellipses, and for a solid retangular section.* The results do not differ much from the above until the ratio of the principal diameters of the ellipse, or of the sides of the rectangle, is greater than 2.

In a ship, the sections are always sufficiently stiffened by frames, beams, and bulkheads, to keep them in shape, and thus allow the torsional shearing stresses to act. Hence, we may conclude that the above result is fairly approximate for the actual case. The fining of the ends of the ship, while diminishing the torsional rigidity, also diminishes the moment of inertia. It will also throw the nodal planes nearer the centre of length. The opposite tendencies of these changes will probably leave the period of torsional vibration comparatively unaffected.

## 10. Torsional Inertia Couples from one Crank and one Pair of Cylinders.

Connecting rod = 9 ft. = 4½ cranks; but, in order to use series already calculated, take the worse case of—

* For the expression of the torsional rigidity in this case, see Love's "Theory of Elasticity," Vol. I. p. 168, equation (69). For numerical values corresponding to ratios of sides from 1·5 to 2·3, see "On Crank Shafts," Table XXIX., by the present writer, Journal of the American Society of Naval Engineers, Vol. IX. p. 354.

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Connecting rod = 4 cranks.

$m$ = moving mass for each cylinder.

$r$ = crank radius in feet.

$\omega$ = angular velocity of crank in radians per second.

$\theta$ = crank angles from top stroke.

T = Turning moment on one crank due to inertia (from two cylinders. Total moving mass = 2 $m$.)

$$T = 2\,m \times \frac{\omega^2 r^2}{g}\,(\cdot 0635 \sin\theta - \cdot 5001 \sin 2\,\theta - \cdot 1920 \sin 3\,\theta - \cdot 0161 \sin 4\,\theta + \cdot 0026 \sin 5\,\theta + \cdot 0004 \sin 6\,\theta + \ldots) \qquad (16)^*$$

C = Torsion couple from cylinder not over crank, due to inertia (= inertia forces × 3 $r$.† Moving mass = $m$).

$$C = -\,3\,r \times \frac{m\,\omega^2 r}{g}\,(\cos\theta + \cdot 2540 \cos 2\,\theta - \cdot 0041 \cos 4\,\theta + \cdot 00007 \cos 6\,\theta + \ldots) \qquad (17)^*$$

The combined couple of second period is—

$$\frac{2\,m\,\omega^2 r^2}{g}\,(\cdot 5001 \sin 2\,\theta + 1\cdot 5 \times \cdot 2540 \cos 2\,\theta)$$

$$= \frac{2\,m\,\omega^2 r^2}{g}\,(\cdot 5001^2 + 1\cdot 5^2 \times \cdot 2540^2)^{\frac{1}{2}} \sin(2\,\theta + a).$$

Hence, the increase of the maximum value of the moment, over the case in which both cylinders are arranged above the crank, is—

$$\frac{(\cdot 5001^2 + 1\cdot 5^2 \times \cdot 2540^2)^{\frac{1}{2}}}{\cdot 5001} = 1\cdot 257.$$

Putting down the increase for all the periods above the first, we have—

| Period. | Increase. | Percentage Increase. |
|---|---|---|
| 2nd | $\frac{(\cdot 5001^2 + 1\cdot 5^2 \times \cdot 2540^2)^{\frac{1}{2}}}{\cdot 5001} = 1\cdot 257$ | 25·7. |
| 3rd | | 0. |
| 4th | $\frac{(\cdot 0161^2 + 1\cdot 5^2 \times \cdot 0041^2)^{\frac{1}{2}}}{\cdot 0161} = 1\cdot 070$ | 7. |
| 5th | | 0. |
| 6th | $\frac{(\cdot 0004^2 + 1\cdot 5^2 \times \cdot 00007^2)^{\frac{1}{2}}}{\cdot 0004} = 1\cdot 034$ | 3·4. |
| 7th | | 0. |
| &c. | &c. | &c. |

Thus the high period torsional couples with which the torsional vibrations of the ship would synchronise are not sensibly increased by the extra couple due to the cylinders being arranged athwartship. Indeed, as $m$ is, on the average of the four

* For equations (16) and (17) see *Engineering*, Vol. LXIV. p. 512, equations (12) and (10); or Journal of the American Society of Naval Engineers, Vol. IX. p. 508, equations (80) and (78).

† In design, crank = 2 ft.; transverse cylinder centres = 5 ft. 10 in. = nearly 3 $r$.

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cylinders, less for this engine than when all the cylinders are arranged over the shaft, the high period couples are actually diminished.

The above deals only with one crank. Whether, for any particular period, the torsional couples from the various cranks will augment or partially neutralise one another will depend on the angles of the cranks. But as in no case do sensible elastic torsional vibrations occur with the ordinary design, we may now be sure they will not occur in the proposed design.

The intensity of the inertia couples decreases very rapidly as we proceed along the series. Thus the ratio of the maximum intensity of the 6th period couple to that of 1st period is, from the above—

$$\frac{\cdot 0004 \times 1\cdot 034}{(1\cdot 5^2 + \cdot 0635^2)^{\frac{1}{2}}} = \frac{1}{3630}.$$

For a comparison, may I be again allowed to refer to the "Inertia Stress of Elastic Gears." From Table X. it is readily calculated that we have for the United States ship *Newark's* low pressure valve gear, supposed rigid—

| | | |
|---|---|---|
| Cut-off | ·76 | ·65 |
| $\frac{\text{Amplitude of 6th period}}{\text{Amplitude of 1st period}}$ | $\frac{1}{245}$ | $\frac{1}{695}$. |
| $\frac{\text{Stress of 6th period}}{\text{Stress of 1st period}}$ | $\left\{ \frac{6^2}{245} = \frac{1}{6\cdot 8} \right.$ | $\frac{6^2}{695} = \frac{1}{19\cdot 3}$. |

At ·65 cut-off the *Newark's* gear worked well. Hence, comparing the latter fractions with the former, we may be confident that a very slight absorption of energy, either from the viscosity of the metal of the ship, from the cargo, or from the water, would effectually damp the elastic torsional vibrations, even if the synchronising period was so low as the 5th or 6th. This, together with the fact that the synchronising period is higher, seems sufficient to completely account for elastic torsional vibrations never having been observed in a steel ship.

## 11. Rigid Body Rotation of Ship from unbalanced Couples.

The 1st period couples present in this engine, but almost absent in the usual design, will be many times too slow to synchronise. Hence, no elastic torsional vibrations will arise. These unbalanced torsional forces can thus only rotate the ship as a rigid body. Let us examine the amplitude of this rigid body vibration.

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First, treat the motion of the piston as simple-harmonic.

D = athwartship distance between cylinders, in feet.
The other letters have the same meaning as in § 10.

The torsional couple of one pair of cylinders (say the high pressure and 2nd intermediate pressure), due to their standing athwartship, is—

$$C = \frac{m\,\omega^2\,r}{g} \cos\theta \times D, \text{ in foot tons.} \tag{18}$$

If I is the moment of inertia of the ship (its displacement being in tons, and radius of gyration in feet); and $\phi$ the angular displacement, torsionally, from the mean position at time $t$—$\phi$ being in radians and $t$ in seconds—we have—

$$C = \frac{I}{g}\frac{d^2\phi}{d\,t^2}. \tag{19}$$

As $\theta = \omega\,t$—

$$\frac{d^2\phi}{d\,t^2} = \omega^2 \frac{d^2\phi}{d\,\theta^2};$$

hence, equation (19) gives—

$$C = \frac{I\,\omega^2}{g}\frac{d^2\phi}{d\,\theta^2}. \tag{20}$$

Equating the two expressions for C given by equations (18) and (20), we get—

$$\frac{I\,\omega^2}{g}\frac{d^2\phi}{d\,\theta^2} = \frac{m\,\omega^2\,r\,D}{g}\cos\theta,$$

or—

$$\frac{d^2\phi}{d\,\theta^2} = \frac{m\,r\,D}{I}\cos\theta. \tag{21}$$

Equation (21) gives at once—

$$\phi = -\frac{m\,r\,D}{I}\cos\theta, \tag{22}$$

and for the maximum value of $\phi$—

$$\phi_1 = \frac{m\,r\,D}{I}. \tag{23}$$

If $b$ is the half breadth of the ship, the linear amplitude of the greatest vibration produced is—

$$\phi_1\,b = \frac{m\,r\,D\,b}{I} \tag{24}$$

This vibration is seen to be constant whatever the speed of engine. In twin screws it will vary, as sometimes the effects from the two engines will be additive and sometimes the reverse.

(Equation (23) could at once have been written down from considering the

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conservation of moment of momentum; but, perhaps, the longer deduction is to be preferred here. As the vibration is so quick, and of such small amplitude, the time of roll of the ship and the action of the water have no perceptible influence, and are not considered.)

Now, from § 4 (*d*) we have—

$m$ for the high pressure and 2nd intermediate pressure cylinders = 6,802 lbs. = 3·04 tons.
$m$ for the 1st intermediate pressure and low pressure cylinders... = 7,556 lbs. = 3·38 tons.

As the cranks are at right angles, it is very readily shown that for the whole of one engine we must take—

$$m = \sqrt{(3{\cdot}04^2 + 3{\cdot}38^2)}.$$

In twin screws the effects of the two engines will at times be exactly additive. Hence we will take—

$$m = 2\sqrt{(3{\cdot}04^2 + 3{\cdot}38^2)} = 9{\cdot}08 \text{ tons.}$$

The United States ship *Alabama** has engines little, if any, greater in power than that in Figs. 2 to 13, Plate LIV., and I will estimate this rigid body vibration for her:—

Displacement (load draught) = 11,562 tons.
Beam, extreme ... ... = 72 ft. 2½ in.

The radius of gyration must be in the neighbourhood of 30 ft.—possibly one or two feet less, but its exact value could be found from the working drawings. Thus, say—

$$I = 11{,}562 \times 30^2.$$

Putting, in equations (23) and (24)—

$$m = 9{\cdot}08,\ r = 2,\ D = \text{(say) } 6, \text{ and } b = \text{(say) } 36 \times 12 \text{ in.},$$

we have—

$$\phi_1 = \frac{9{\cdot}08 \times 2 \times 6}{11{,}562 \times 30^2} = 1{\cdot}05 \times 10^{-5},$$

$$\phi_1 b = 1{\cdot}05 \times 10^{-5} \times 36 \times 12 = {\cdot}0045 \text{ in.}$$

That is an amplitude of less than $\frac{1}{200}$ of an inch, and the total movement is less than $\frac{1}{100}$ of an inch.

Aiming a gun broadside at a target three miles distant, when the engines are going, the total apparent movement of the sight on the target will be—

$$\frac{1}{100} \times \frac{3 \times 5{,}280}{36} = 4{\cdot}4 \text{ in.}$$

a quantity quite invisible.

Treating now the effects of the shorter period couples,—

If the speed of engine is doubled, the 1st period vibration will, as we have seen, remain of the same amplitude, though the couple producing it is increased four times.

* See Journal of the American Society of Naval Engineers, Vol. XII. p. 855.

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Hence, for the 2nd period couple to produce as large an effect as the first, its value would have to be 4 C. As it is actually less than $\frac{1}{4}$ C, the effect $\phi_2$ *from each crank* for 2nd period is less than $\frac{1}{16}\,\phi_1$ from each crank for 1st period. But, as 1st period couples change to opposite phase every half revolution, those of 2nd period change every quarter revolution. Thus, for two cranks at right angles, the effects of 2nd period exactly oppose one another; and the resultant effect is merely due to the difference of mass actuated by each crank. Therefore, $\phi_2$, *from the whole engine*, for 2nd period, is a small fraction of $\frac{1}{16}\,\phi_1$, from the whole engine, for 1st period.

Similarly, it can be shown that $\phi_3$, $\phi_4$, &c., for all high-period forces are so small as to be quite insensible.

In the tandem design, Fig. 1, page 300, these unbalanced inertia couples scarcely exist. The same is true of that form of the engine, included in the patent, in which three cylinders stand in line athwart the shaft; but this is not suitable for shipboard.

If we compare two ships in which all the linear dimensions of the ship and machinery are changed in the same proportion, we have $r$, D, and $b$ changing as the first power, $m$ as the third power, and I as the fifth power of the linear dimensions. Therefore the numerator of equation (24) would change as the sixth, and the denominator as the fifth power; and, therefore, the linear amplitude $\phi_1\,b$ would be smaller in the same proportion as the ship was smaller. This reduction would, however, leave the smaller ship stronger than it would usually be made, and the reduction of $\phi_1\,b$ would actually be somewhat slower.

But it is evident from this that there is a large margin left for giving the fuller power necessary in very fast small craft, without objectionably increasing $\phi_1\,b$. For it will be noted that the heating and grate surfaces vary as the second power of the linear dimensions; and so also do the horse-power (for constant piston speed) and the wetted surface. So that the reduction supposed leaves the same horse-power per square foot of wetted surface. The linear amplitude $\phi_1\,b$ would certainly have to be increased a very considerable number of times over that just calculated for the *Alabama*, before it became troublesome. And I doubt if any ship could carry machinery of the proposed design in which this vibration would be more than minute. It is a great advantage that the exact result can so easily be predicted in any case.

Indeed, we have a much worse case in twin screws with unbalanced engines, for the leverage is far larger and the unbalanced forces frequently great. But, usually, there is no noticeable effect, and, I think, never one of serious magnitude.



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## DISCUSSION.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Gentlemen, I am sure we all thank Mr. Macalpine for his paper. Properly speaking our time is now up, but Mr. Yarrow wishes to say a few words.

Mr. A. F. YARROW (Vice-President): My Lord and Gentlemen, I should like to take the opportunity of congratulating Mr. Macalpine on this very interesting contribution to the vibration problem. It will be a valuable addition to what has gone before. Should there be any points in the paper which I might criticise, I would, with your lordship's permission, put them in writing and send them to the Secretary.

The PRESIDENT: I have Mr. Macfarlane Gray's name down as wishing to speak. I do not know whether he thinks he could do it as well by sending his remarks in writing to the Secretary.

Mr. J. MACFARLANE GRAY (Member of Council): My Lord, I could do as you suggest had I any remarks to make, but I only wish to say that I have known Mr. Macalpine for the last fifteen years. He was introduced to me by the late Dr. Kirk as a practical man in whom he had great confidence, and ever since then, I have watched his progress with admiration. I have done something in the way of simplifying balancing, but I simply surfaced the thing over, and when I came to a difficulty I tried to just smooth it over for many of our non-mathematical members; but Mr. Macalpine never skips anything. If he comes to a difficulty he seems to glory in it, and he goes to the very bottom of it. The younger members especially have in Mr. Macalpine's paper a capital exercise; by carefully studying it they may improve themselves. I have great pleasure in saying that I have been much interested in the paper, and I wish Mr. Macalpine success in every way.

The following contributions were received after the close of the Meetings:—

Mr. A. E. SEATON (Member): I regret that I am unable to join in the discussion on Mr. McAlpine's paper; I may, however, say that the engine he proposes is identical with one designed by John Elder for compound engines of large size (where two low-pressure cylinders are desirable) in the late sixties of last century. I understood at the time that it was patented, but it may not have been, for in 1870 C. & W. Earle made a set of these, with cylinders, I think, 36 in. and 72 in. by 42 in., and fitted them in the s.s. *Canopus*, of the Moss Line, Liverpool. They were not a success, and they were removed after only three or four years' service, having been a very costly experiment for both builders and owners. The design is radically wrong from a construction point of view, inasmuch as the reaction from one pair of cylinders is not taken direct to the crank shaft, but practically is borne by the ship, and at best transmitted to the shaft through the ship's bottom framing. Time would soon show that this defect was a bad one, however well the engine was bolted down. Then, too, the beams or weigh-levers gave great trouble in the *Canopus*, running at 50 revolutions: at 112 revolutions, we may presume, these troubles would be magnified. Then, too, although there are four engines, you have only the qualities of a two-crank, except for the balancing. Now I don't want to throw cold water on anyone's schemes, nor do I under-estimate the advantage of the absence of vibration, but I cannot shut my eyes to the price paid for this luxury, especially when it takes the form of a sacrifice of those principles which go to make a sweet-running and economic engine. We have yet to find out what the public will pay for non-vibration, and, until we do, we must be cautious

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of the cost of obtaining it. I may add that about 1880 we removed the four cylinders from the engines of a Cardiff steamer of small size (the engines being of identical design with Mr. McAlpine's), and replaced them with the orthodox two-compound, the owners finding the consumption of fuel was so heavy and the speed so poor, compared with similar ships having ordinary engines.

Mr. Billetop (Member): I have read Mr. McAlpine's paper with great interest, the more so as I am aware of the exceptional opportunities he has had for, and the time he has devoted to, the study of the problem of the vibrations of marine engines. In an engine of the design proposed by Mr. McAlpine, the vibration, judging from the results of experiments I have already made in this direction, should be practically *nil*. I have tested small engines, built on the same principle as Mr. McAlpine suggests, though differing somewhat in general design, and have found such engines to be quite steady when running at any speed up to 2,000 revolutions per minute. During the tests, the engines were not held down by bolts or fixtures of any kind, and a glass filled with water, and placed on top of the cylinders, did not show the slightest surface tremor when running under those conditions. I have already discussed the constructional and other practical points with Mr. McAlpine, much more fully and thoroughly than could well be done within the limits of his paper, and I am convinced that the points which appear at first sight to be objectionable or difficult could easily be met or overcome. I have also looked into the arrangement of such engines in ships of the destroyer type, as I thought there might be some difficulty in fitting twin-engines of this type in ships of narrow beam. There seems, so far as I can see, no difficulty on this point, and I find that a set of engines after Mr. McAlpine's design can easily be fitted within the same engine-room space as the present types of engines occupy.

Mr. Philip Watts, F.R.S. (Member of Council): I regret I was unable to attend the reading of Mr. Macalpine's paper, but I have read it through with great interest. I have had his proposals before me for some time, and had his calculations carefully verified, and have satisfied myself of their accuracy. His engine appears to me thoroughly practicable, and to possess advantages over other engines.

Mr. W. C. Borrowman (Member): The subject of this paper has lately become one of immense importance in marine engineering, and has occupied the attention of most of the eminent engineers of the country. Some have approached the problem in a purely theoretical direction, and have contented themselves by pointing out what a perfectly balanced system requires, while others have met the difficulty in a practical way by applying, more or less haphazard, counterbalance weights to the revolving parts, and have thus mitigated to a certain extent the evils of vibration. Both have been useful, but neither has overcome the difficulty. The two most practical solutions of the problem are those of the Parsons turbine and the Yarrow, Schlick, and Tweedy system. The former is, of course, a special case, and does not come under the present consideration of the question, and the latter has till now been accepted as the most feasible way out of the difficulty. As a matter of fact, however, it is not by any means an ideal solution either theoretically or practically, as anyone can prove who has adopted it. This is not said by way of disparaging Messrs. Yarrow, Schlick, and Tweedie's method, but merely to point out that, in so far as vibration of marine engines is concerned, there has not hitherto been a perfect means of meeting it. They have all been, more or less, simply an application of the old counterbalancing methods, with the result that vibration is still experienced. It is perhaps hardly correct, while discussing a proposed system of balancing, to criticise any other, but the Yarrow,

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Schlick, and Tweedy arrangement is so well known, and has been all but universally adopted as the means of meeting vibration, that one cannot entirely eliminate it from one's mind when confronted with what is virtually a counterblast to it and all others. The Macalpine system claims, in a most emphatic way, to be the only solution of the problem, not only theoretically, but practically as well, and this is what everyone has been looking for. Judging the two systems theoretically in the first instance, as a merely mathematical solution, and without in any way accepting the Macalpine system as being perfect, one has to admit that it is certainly a more nearly correct one than the others; and that, barring a few very infinitesimal conditions incident to all mechanisms, to all intents and purposes it meets the case. It does not always follow, however, that what holds good in a mere question of equations, holds equally good when applied to material, and a system which claims perfection as its characteristic must satisfy very severe tests in both directions; and, when looked at from a practical standpoint, it has to be confessed at the first glance that there are features in the Macalpine design antagonistic to one's ideas of practicability. Mr. Macalpine himself has evidently felt this, and has taken no small pains to show how groundless are any objections raised to the application of his system. He criticises his own design very fairly and impartially, and anticipates in a thoroughly practical way nearly every objection that may be raised against his arrangement. If we take as the latest development of marine practice the now common quadruple-expansion, four-crank type of engine, as designed to satisfy the requirements of balancing, it does not compare favourably with the proposed one. The space occupied by such machinery in a fore and aft direction is very considerable, and in some cases prohibitive; there is the objectionable feature of having the first and second cylinders at the extreme ends of the engine, thus necessitating a long exposed exhaust pipe between them; and the occasional serious loss of efficiency due to the consequent drop in the steam pressure. There is the further question of weight, and the consequent one of price. In the proposed engine, none of these charges are applicable; in fact, they are avoided in a very simple and practical manner, and, apart from the question of vibration altogether, the design leads to a simplification of parts, all easily got at for attention, easily disconnected and removed for overhaul, and not any heavier than their counterparts in the ordinary marine engine. To the owner there is the further decided advantage of having less space occupied and less permanent deadweight to be carried, thus adding to the space and weight possible in the shape of cargo and the consequent carrying power of the vessel. A rough calculation of the weight saved shows this to be from 18 to 23 per cent. Perhaps the least happy feature of the design is the lever employed for transmitting the power of one cylinder to the crank. This is, however, a necessary feature in the question of balance, and, apart from the natural prejudice against innovation, it forms no insurmountable difficulty nor very serious objection. The idea of a lever is not novel, as it has been and is employed in some very successful engines now at work, and is still a feature of the highest class marine engines of the present day. If care is exercised in its design, and ample surface given to its bearings, there need be no anxiety as to its working properly. Even with this lever included, there are fewer parts demanding attention in the proposed design than in a quadruple engine of the usual type; and, as stated, with a little exercise of care in designing, all the difficulties and objections which at first suggest themselves become avoidable. It is an engine which should commend itself for all classes of vessels—naval, passenger, and cargo.

Professor W. E. Dalby, M.A., B.Sc. (Associate): With reference to Mr. Macalpine's remark on page 300, that "with five and six-crank engines we can balance for first, second, and fourth period forces, the arrangement being the worst possible for the sixth period," it might be noticed that the

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"worst possible" only represents a force of the maximum value of 4·1 lbs. weight for a set of reciprocating parts weighing five tons, the speed being 88 revolutions per minute. This is calculated from the formula No. 1 given by Mr. Macalpine on page 298. If six such sets of masses were used in a six-crank engine, the unbalanced force due to the motion of the whole thirty tons of reciprocating parts is only 24·6 lbs. weight, for an engine arranged in the way shown in my paper of last spring. Suppose the six cylinders arranged in the way Mr. Macalpine proposes at 7 ft. centres, this 24·6 lbs. might certainly be eliminated in the vertical plane, but only by introducing a periodic torsional couple, whose maximum value is, roughly, 175 foot-tons for each pair of cylinders athwart ship, in which the reciprocating parts weigh 5 tons per cylinder. Whether a ship, engined in the way Mr. Macalpine suggests, would run steadier than one engined with a six-crank engine of equal power, is a nice question. It would be interesting to have a practical trial of the engines proposed, and I hope Mr. Macalpine will soon be successful in placing such an engine in a ship.

Mr. J. H. Macalpine (Member): I have to thank the gentlemen who have so kindly contributed to the discussion of my paper. Mr. Seaton's statements have been discussed by me in *Engineering*, of July 12 and 26, more fully than it is necessary or desirable to do here. I will only partly reproduce my first letter. I brought forward the strongest evidence of one who had much to do with the *Canopus*, and whose opinion has great weight, to show that the whole reason for the failure of her engines was bad design. His opinion, from experience with these engines, is that they would have been perfectly satisfactory if properly proportioned. The error of Mr. Seaton's statement regarding the stresses that are "transmitted to the shaft through the ship's bottom framing" is so evident that I need not again refute it. On account of extra cost and complication, and the greater space occupied, the four-crank engine was introduced very reluctantly. Its only recommendation was that it offered a rather better, though far from perfect, balance. What other desirable qualities, besides partial balance, a four-crank engine possesses over a smaller number of cranks, I asked Mr. Seaton to say, but he has not yet done so. He admits that I have attained the desired balance with a two-crank engine—that is, by simplification and without extra cost—and yet he condemns it! Instead of any sacrifice of principles, surely the absence of vibration is one of the most essential elements in making "a sweet-running and economic engine." The vibration which is admittedly so very desirable to avoid in the ship must surely have a detrimental effect on the engine, which must be vibrating if the ship is. We may not know how much the public would pay for non-vibration; but, judging from the deep interest taken in the problem for many years, not only within the profession, but by all who have had to use fast steamers, we know that a satisfactory solution of it would be very highly valued. Mr. Seaton's last paragraph is not entirely intelligible. Putting aside very slight differences of friction in the mechanism—in which, I think, it can be demonstrated that my proposed engine would compare favourably with the ordinary four-crank—economy of steam depends on the action of steam in the cylinders and passages. Either the steam has some special predilection for cylinders arranged over the crank, or the engine Mr. Seaton instances was badly designed or in bad repair. The development of the allied subjects of steamship vibration and complete balancing is very recent. Such considerations formed no part of the objects of the design in the engines Mr. Seaton instances. Hence they are no anticipation of what I have brought forward in my paper. I have before me, as I write, a photograph of the engine Mr. Billetop refers to, and it may be of interest to note that even the small surface marks on the metal come out quite sharply, though the engine was running at 2,000 revolutions when photographed. This shows in a most convincing manner how steady it was. It was the type of engine I referred to in the first

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paragraph of § 3. I was, of course, aware of the smallness of the forces noted by Professor Dalby, which are given by equation (1). These forces are, however, calculated on the supposition of absolutely no elastic response from the ship, or engine foundation. They are due to an extremely small amplitude of vibration. If the elastic response was of the same order of magnitude, it would induce practically no absorption of energy either in the ship or surrounding water; that is, so small an amplitude would not produce frictional sliding of the parts in contact in the ship, or sensible dissipation of energy in the water, either from viscous flow or wave making. Hence, under favourable circumstances, we would expect an elastic response many times larger in amplitude than the original vibration, before communication of energy from the engines and its absorption in the ship and surrounding medium would reach equality, thus preventing further growth of the elastic vibration. The inertia forces would, of course, rise in proportion to the amplitude. This is what I had in mind when discussing, in *Engineering* (Vol. LXVII. page 54), Professor Dalby's excellent paper of 1899, read before this Institution. After noticing this communication and absorption of energy, I conclude, "Hence, we sometimes have considerable vibration produced by almost incredibly small forces." Vibrations of sixth and even higher periods have been noted, and a very short period vibration need only be of very minute amplitude to cause great discomfort. As an example of a very rapid vibration arising from very small forces, I may note one cited by me in the same letter in *Engineering*. The electric light engine on board Mr. J. Pierpont Morgan's yacht *Corsair* ran about 550 revolutions per minute, and caused a most disagreeable vibration, principally noticeable at the stern. The first and second period vibrations were about equally marked. The second period forces, recurring at the rate of 1,100 per minute, could not have been very greatly in excess of the value given by Professor Dalby. The first period vibration was very distinctly felt as a back and forward movement of the part touched, while that of second period seemed to be the one which gave rise to a peculiar tingling feeling through the fingers, or other parts of the body, touching bare woodwork, iron, or any hard surface. Though I think there is a distinct risk of disagreeable vibration in some cases with five and six-crank, for the reasons just given, I do not doubt that in probably the majority of cases, they would run with great smoothness. By far my strongest objection to them is expressed in the sentence after that which Professor Dalby quotes, "But the engine is complex and costly, and takes up much valuable space in the ship"; the "but" referring, not to the unbalance of sixth period, but to the very exact balance attained. I give expression to the same opinion both in the letter to *Engineering* in 1899, above referred to, and again in the discussion of Professor Dalby's spring paper. In comparing the large torsional couple I introduce, with the small sixth period force, it must be recollected that the former is of first period, and consequently that the admissible amplitude of vibration from it is very many times greater than that allowable in a vibration of sixth period. Indeed a vibration of sixth period which would be absolutely intolerable would, if reduced to first period, require to have its amplitude very much increased before it could cause any annoyance whatever. I think this is the clear teaching of experience. Again, if the ship responds to sixth period forces, it responds as an elastic body, whereas its response to the large torsional couple is as a rigid body. In the first case, the amplitude can be very largely increased by accumulation of energy of vibration; in the second case, this cannot happen. The points Professor Dalby raises are of the deepest interest and importance. While thanking him for bringing them forward, I must apologise for the length of reply they have led me to give.

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# STANDARDISING THE RESULTS OF SHIP CALCULATIONS.

By Professor J. H. Biles, Member of Council.

[Read at the Summer Meetings of the Forty-second Session of the Institution of Naval Architects, June 27, 1901; the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D., President, in the Chair.]

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The object of this paper is to call the attention of naval architects to the desirability of recording the results of each one's calculations in a manner that will make them directly comparable with everyone else's. At the present some are directly comparable, such as co-efficient of fineness, speed co-efficients, $\frac{\Delta^{\frac{2}{3}} \times S}{\text{I.H.P.}}$, &c., but many are not. For instance, curves of stability are set off generally upon arbitrary scales, so that there is no apparent relation between the righting arm and the size of the ship. Bending moment calculations are commonly dealt with in the same isolated manner. The work done for one ship is of very little use for any other. This paper is written with a view to calling the attention of naval architects to this subject, and also to suggest methods of standardising the results of some calculations so that they may be here discussed, and afterwards experimented upon in practice, in either their present form, or in such modified form as the judgment of members may suggest.

Let us take first the displacement paper. A form is shown in Table A (Plate LVI.) which the author devised many years since for obtaining by Simpson's rules the displacement of a set of lines at a series of draughts. This form is given as showing one method of arranging such work, and to illustrate the general question under consideration. The table enables the calculator to obtain displacement, vertical and horizontal position of centres of buoyancy, areas of water lines, and mid sections, for a series of draughts. If these results be set off in terms of draught, we shall have for the particular form we are dealing with curves of displacement, locus of vertical and longitudinal centres of buoyancy, water line, and mid section areas. Such a series of results is shown in Fig. 1.

It is usual to set off these results upon some arbitrarily chosen scale, such as 1 in. = 1,000 tons for displacement, or $\frac{1}{2}$ in. = 1 ft. for vertical and longitudinal measurements of centres of buoyancy. These curves convey nothing to the eye, except the variation of result in terms of draught. To make any use of them, the number of inches of their ordinates must be measured, and the result in inches must be multiplied

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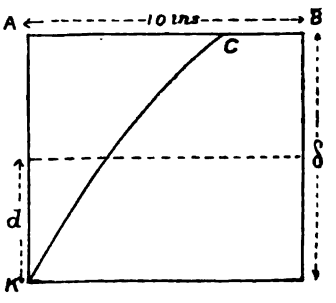
by the scale upon which they have been set off. When this is done, we know the value of the particular ordinate we have been dealing with, and what it means in relation to the particular set of lines from which it was taken. Its usefulness stops at this point. If we have similar results for other sets of lines their usefulness is similarly limited. Each set of results is isolated when it is set off upon such scales as are usual.

Suppose, however, we adopt a method of setting off results, so that we can remove this isolation, and make each ship directly comparable in all these respects with all other ships. We shall then be able to compare all our own ships with each other, and, if other naval architects are willing, with all theirs also.

The results we have been discussing are due to form and dimensions. The system we shall now consider is one in which form only is taken into account, dimensions being altogether relegated to the domain of "practical application."

Suppose we have a vessel whose dimensions are L, B, $d$ (length, breadth, and draught), $\delta$ being the load draught. The product $\frac{L B \delta}{35}$ will give us the number of tons of salt water which the circumscribing parallelopipedon, or, as it is perhaps more commonly known, "the block" would displace. Suppose that for all ships, whatever their forms or dimensions, we let the displacement of this block be represented by a fixed length, say, 10 in. (or, in those countries which are blessed with a metric system, say, $\frac{1}{4}$ of a metre, = 9·84 in.). Whatever the displacement at any draught in any *form* may be, it will have a definite relation to the volume of the circumscribing block, and this relation will remain absolutely unaltered, however we may alter the individual values of L, B, and $d$, provided that we alter all measurements in the directions in which L, B, and $d$ respectively are measured, in exactly the same ratio in which L, B, and $d$ have been altered. In other words, one of the qualities which remains unchanged for all variations of L, B, and $d$, when form characteristics are maintained, is the block co-efficient. A block has a co-efficient of unity, and this may be taken as our unit of value for displacement. We always consciously or unconsciously fall back upon this value as a unit in measuring fulness; why should we not begin with it?




If we agree to let 10 in. = $\frac{L B \delta}{35}$, and at the same time let unity represent the volume of the block, we shall have 10 in. = (say, A B) = 1 of block co-efficient, and 1 in. = ·1 of block co-efficient. But 10 in. = $\frac{L B \delta}{35}$ tons, and 1 in. = $\frac{L B \delta}{350}$ tons. If we set off the actual displacement at the load water line of the ship we are considering on the scale of 1 in. = $\frac{L B \delta}{350}$ tons (say, A B), then the ratio of the

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number of inches which will represent the displacement to 10 in. will be the block co-efficient of the set of lines under consideration.

Suppose we set down $\delta = AK = 10$ in. Then the draught at any intermediate point can be set up from K on a scale 1 in. $= \frac{\delta}{10}$ ft. of draught. If the values of the displacement at any other draught $d$ be set off on the scale, 1 in. $= \frac{LB\delta}{350}$ tons at the corresponding draughts (which will be on a scale 1 in. $= \frac{\delta}{10}$ ft.), we can get a curve K C which will show the variation of displacement in terms of draught, but will also show the displacement at any draught for any ship of similar form whose dimensions are known, upon a scale whose absolute value can be determined by substituting the particular values of L, B, and $\delta$ in the equation 1 in. $= \frac{LB\delta}{350}$ tons. Thus, by setting off the result of *one* ship's calculation in this suggested manner, we have removed the isolation in which it stood, and have made its results immediately applicable to an indefinite number of cases of differing dimensions. The only quality which we have retained constant is the character of the form. If we work out the results for another form and set them off in the same way, the comparison of the results due to difference of form will be at once apparent to the eye. All question of difference of dimension will be absolutely eliminated, and we shall have a comparison of the results of form only. In this way the result has been standardised. I know of no more suitable word to use, or I would adopt it.

Displacement has been chosen merely to illustrate the principle of standardisation. The unit length was chosen at 10 in. as a matter of convenience. Whatever length is chosen should be adopted by all who use this system, partly for purposes of direct comparison by placing tracings of the results of different forms above each other; and partly because the eye can judge immediately of the qualities if the absolute scale is always the same.

Fig. 2 (Plate LVIII.) shows a considerable number of forms, the displacement curves of which have been standardised. The dimensions of the actual ships from which they have been deduced are given, and also the types of vessels are stated. It will be seen that the comparison of form characteristics, as far as displacement is concerned, is shown for a very wide range of types.

In a similar manner the results of calculations of water-line areas, midship section areas, longitudinal and vertical positions of centre of buoyancy, heights of metacentre above centre of buoyancy, may all be set off. Such results are shown in Figs. 3 to 7 (Plates LVIII. and LIX.) respectively.



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The scales upon which the actual results have been set off are as follow :—

Fig. 3. Water-line area ... ... ... ... ... 10 in. = L B sq. ft.
1 in. = $\frac{L B}{10}$ " "

Fig. 4. Midship section area ... ... ... ... 10 in. = B $\delta$ " "
1 in. = $\frac{B \delta}{10}$ " "

Fig. 5. Distance of centre of buoyancy from middle ordinate = $l$.
If 100 in. = L, and we set off $l$ on the same scale, it will be on scale of 1 in. = $\frac{L}{100}$ ft., and is plotted from the centre line of the diagram, the after side being to the left.

Fig. 6. Distance of centre of buoyancy above keel = K B = $b$.
If 10 in. = $\delta$, and we set off $b$ in this scale, it will be on scale 1 in. = $\frac{\delta}{10}$.

Fig. 7. Height of metacentre above centre of buoyancy = B M.
The transverse B M of a block is = $\frac{I}{V} = \frac{L B^3}{12 L B d} = \frac{B^2}{12 d}$.
If 10 in. = $\frac{B^2}{12 \delta}$, 1 in. = $\frac{B^2}{120 \delta}$.

If we set off actual B M's on this scale at the corresponding draughts (on a scale of 1 in. = $\frac{\delta}{10}$ ft.), we shall get standardised B M curves, such as are shown in Fig. 7 (Plate LIX.). Longitudinal B M curves may be set off in a similar manner.

These results are perhaps not so valuable as those which give the actual vertical positions of the metacentre in the ship, because it is generally its relation to the centre of gravity which is of most interest.

To find this, it is necessary, as usual, to set up M in two operations, first by setting off K B, and then setting B M above it. If both K B and B M be set off on the scales already used, viz., 1 in. = $\frac{\delta}{10}$ ft., and 1 in. = $\frac{B^2}{120 \delta}$ ft., the resulting curve of K M will, especially in the lighter draught parts of the curve, be outside the scope of the 10 in. diagram. If, however, we note that the K M of a block is $\frac{\delta}{2} + \frac{B^2}{12 \delta}$, and let 5 in. = $\frac{\delta}{2} + \frac{B^2}{12 \delta}$, then K M's can be set off on the scale 1 in. = $\frac{\delta}{10} + \frac{B^2}{60 \delta}$.

The resulting curves of K M show the position of metacentre for each form in terms of the K M of a block. These curves are shown in Fig. 8.

Ratio curves can also be set off by letting 10″ represent a co-efficient of unity, so that 1″ = ·1 of co-efficient. Curves of block, prismatic, midship area and water-line area co-efficients, in terms of draught, are shown in Figs. 9, 10, 11, and 3 (Plates LX. and LVIII.) respectively. The reason why the water-line areas co-efficient coincides with the water-line areas is that the draught is not involved in the calculation of the co-efficient, and B remains constant.

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The mid-section areas could easily be determined from the co-efficient of mid-section areas by measuring the ordinate (at draught $d$, say) to the scale $1'' = \frac{B\,d}{10}$. The corresponding ordinate at draught $d$ on the mid-section area curve is set up to scale $\frac{B\,\delta}{10}$ ($\delta$ being load draught).

The scales corresponding to the different diagrams are given in the following table :—

CURVES ON 10″ DIAGRAMS.

| No. of Fig. | — | Vertical Scale. | Horizontal Scale. |
|---|---|---|---|
| 2 | Displacements ... ... ... | $1'' = \frac{\delta}{10}$ (draught scale) | $1'' = \frac{L\,B\,\delta}{350}$ |
| 3 | Areas water-line planes ... | $1'' = \frac{\delta}{10}$ " " | $1'' = \frac{L\,B}{10}$ |
| 3 | Co-efficient water-line planes... | $1'' = \frac{\delta}{10}$ " " | $1'' = \cdot 1$ of co-efft., *i.e.*, $\frac{L\,B}{10}$ |
| 4 | Areas mid-sections ... ... | $1'' = \frac{\delta}{10}$ " " | $1'' = \frac{B\,\delta}{10}$ |
| 5 | Longitudinal centres of buoyancy | $1'' = \frac{\delta}{10}$ " " | $1'' = \frac{L}{100}$ |
| 6 | Centres of buoyancy, K B's ... | $1'' = \frac{\delta}{10}$ | $1'' = \frac{\delta}{10}$ (draught scale) |
| 7 | Transverse metacentres, B M's | $1'' = \frac{B^2}{120\,\delta}$ | $1'' = \frac{\delta}{10}$ " " |
| 8 | " " K M's | $1'' = \frac{\delta}{10} + \frac{B^2}{60\,\delta}$ | $1'' = \frac{\delta}{10}$ " " |
| 9 | Block co-efficient ... ... | $1'' = \frac{\delta}{10}$ (draught scale) | $1'' = \cdot 1$ of co-efft., *i.e.*, $\frac{L\,B\,d}{350}$ |
| 10 | Prismatic " ... ... | $1'' = \frac{\delta}{10}$ " " | $1'' = \cdot 1$ of co-efft., *i.e.*, $\frac{M \times L}{350}$ |
| 11 | Co-efficient of mid-section ... | $1'' = \frac{\delta}{10}$ " " | $1'' = \cdot 1$ of co-efft., *i.e.*, $\frac{B\,d}{10}$ |

NOTE.— $\delta$ = load draught.
$d$ = general draught.
L = length between perpendiculars.
B = breadth moulded.

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Stability curves may be treated in a similar manner. Suppose we have a curve of centres of buoyancy for a form inclined at some angle $\theta$ for all displacements, from the total volume or bulk of the form to a zero of displacement.

H (Fig. 12, Plate LX.), the centre of bulk, will be one extremity of this curve, and T, the tangent to the midship section at an angle $\theta$, will be the other. Suppose D is the depth to the upper deck at side, and B is the extreme breadth amidships. Suppose the co-ordinates of any point $b$ on the curve to be $x$ $y$. If we change the dimensions of the form to $B_1$ and $D_1$, we can immediately find the value of the co-ordinates of the point corresponding to $b$ in the new form; call it $b_1$. Suppose $x_1$ $y_1$ to be co-ordinates of $b_1$, then—

$$x_1 = \frac{B_1}{B} x ; \qquad y_1 = \frac{D_1}{D} y ;$$

but $x_1$, $y_1$ will be the co-ordinates of a point of the centre of buoyancy curve in the new form at an angle $\theta_1$,* such that—

$$\text{Tan } \theta_1 = \frac{D_1 B}{D B_1} \times \tan \theta. \dagger$$

---

* $\theta_1$ may be called the "corresponding angle" to $\theta$.

† Suppose K $b$ to be a line through the keel making $\theta$ in the original form, and $b$ F to be the vertical, cutting the horizontal through the base at F. In the new form, where beam is expanded from B to $B_1$ and depth from D to $D_1$, $b$ becomes $b_1$ and F becomes $F_1$ such that—

$$b_1 F_1 = \frac{D_1}{D} . b . F.$$

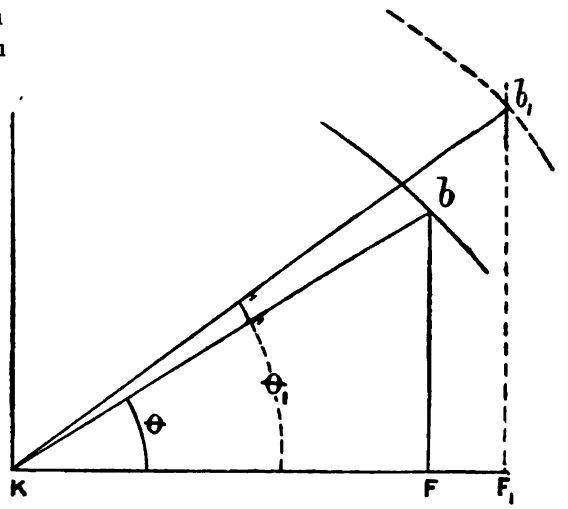
$$K F_1 = \frac{B_1}{B} K F.$$

$$\frac{b_1 F_1}{K F_1} = \tan \theta_1.$$

$$\frac{b F}{K F} = \tan \theta.$$

$$\text{Tan } \theta_1 = \frac{D_1 B}{D B_1} \frac{b F}{K F}.$$

$$\text{Tan } \theta_1 = \frac{D_1 B}{D B_1} \text{Tan } \theta.$$

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If D = 10 in. and B = 10 in.—

$$\text{Tan } \theta_1 = \frac{B}{D} \tan \theta.$$

If in this form, whose breadth and depth are each 10 in., we set off a series of values $x_1$ and $y_1$, corresponding to a series of values of $x\,y$, such that—

$$x_1 = \frac{10}{B}\,x \text{ in.}; \qquad y_1 = \frac{10}{D}\,y \text{ in.},$$

$x$, $y$, B and D being in the same units, we shall have a curve at an angle $\theta_1$ (where $\tan \theta_1 = \frac{B}{D} \tan \theta$), which will show the characteristics of the form as far as they relate to the position of centre of buoyancy on a cross curve of stability.

If the cross curves for all forms are brought to this scale, we shall have a means of comparing directly the effect of form upon the centre of buoyancy.

It is obvious that the reverse of this process will give us a cross curve of stability for a vessel of any chosen values of B and D; so that, if we have a curve for any one set of dimensions, we can obtain it for any other at the corresponding angle.

It is evident that we may set off the lines of a ship, that is, the form itself, in a similar manner. The breadth and draught ordinate may each be set off respectively on scales such that 10 in. = B, or 1 in. = $\frac{B}{10}$ ft., and 10 in. = $\delta$, or 1 in. = $\frac{\delta}{10}$ ft.

We may also set off length ordinates such that, say, 100 in. = L ft., or 1 in. = $\frac{L}{100}$ ft. All forms would by this means be brought to a directly comparable scale.

Other methods of comparing lines are given which have other advantages than this method. For instance, in Fig. 13 (Plate LXI.) the length is on a scale of 1 in. = $\frac{L}{40}$ ft., and the transverse sectional areas are plotted to a scale of 5 in. = B $\delta$ sq. ft., or 1 in. = $\frac{B\,\delta}{5}$ sq. ft. These curves show the longitudinal distribution of displacement.

It is suggested that this method does not measure the fineness quite so well as that shown in Fig. 14 (Plate LXI.), in which the length is set off on a scale of 1 in. = $\frac{L}{40}$ ft., and the ordinates are the $\frac{\text{area of half cross section}}{\text{draught}}$ set off on the same scale as the length. This ordinate is really the mean half breadth of the section, and, as the draught is assumed constant for each line, it is really a cross sectional area curve, the same as in

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Fig. 13, but to another scale, and one which is directly comparable by the eye with the length. The angles of the lines at the ends are better measures of the fineness than in Fig. 13.

To standardise the results of strength calculations, the following method is suggested. Attention has already been called to it by the author in a paper on "The Strength of Large Ships," read before the Institution of Engineers and Shipbuilders in Scotland in 1893. Taking the length of the ship in all cases to be 20 in., the longitudinal scale will be 1 in. $= \frac{L}{20}$ ft., where L = the length of the vessel in feet. The longitudinal distribution of buoyancy is shown by a curve, set off so that its mean ordinate is always 3 in., and in consequence the area of the buoyancy curve will be $3 \times 20 = 60$ sq. in. This area will represent the total buoyancy or displacement, and, therefore, 60 sq. in. $= \Delta$, the displacement, and 1 sq. in. $= \frac{\Delta}{60}$ tons. The buoyancy curve is set off on such a scale that the mean ordinate of it is 3 in. The mean ordinate is $\frac{\Delta}{L}$, and consequently 3 in. $= \frac{\Delta}{L}$ tons per foot of length, *i.e.*, 1 in. $= \frac{\Delta}{3L}$ tons per foot of length.

All the ordinates of the weight and buoyancy curves should be set off on this scale, and the area of the resulting difference or load curve be read off, remembering that each square inch of area represents $\frac{\Delta}{60}$ tons. The area of the load curve, taken from point to point, gives the shearing force curve, and for convenience the number of square inches of this load curve is set off on a scale such that 2 sq. in. of load curve = 1 linear inch of shearing force curve. But 1 sq. in. of load curve, as shown above, is equal to $\frac{\Delta}{60}$ tons, therefore 2 sq. in. $= \frac{\Delta}{30}$ tons. Hence the scale for the ordinate of the shearing force curve is 1 in. $= \frac{\Delta}{30}$ tons.

The bending moment curve is obtained by the integration of the shearing force curve. The scale chosen in this case is such that 3 sq. in. of shearing force curve = 1 linear inch of bending moment curve. But 1 sq. in. of shearing force curve is equal to—

$$\frac{\Delta}{30} \times \frac{L}{20} = \frac{\Delta \times L}{600} \text{ foot tons.}$$

Hence—

$$3 \text{ sq. in.} = \frac{\Delta \times L}{200} \text{ foot tons;}$$

that is, the scale for the ordinate of the bending moment curve is—

$$1 \text{ in.} = \frac{\Delta \times L}{200}.$$

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To summarise the results :—

$$1 \text{ in. of length} = \frac{L}{20} \text{ feet.}$$

$$1 \text{ in. of weight and buoyancy} = \frac{\Delta}{3\,L} \text{ tons per foot.}$$

$$1 \text{ in. of shearing force} = \frac{\Delta}{30} \text{ tons.}$$

$$1 \text{ in. of bending moment} = \frac{\Delta \times L}{200} \text{ foot tons.}$$

The above scales have been chosen in a purely arbitrary manner, but the reason for taking the figures given is that the diagram in all cases may come within the scope of specially prepared sheets of paper 20 in. by 20 in. which are divided decimally.

The advantage of the above method lies in the fact that, irrespective of size, the height of the bending moment curve gives at a glance the *intensity* of the bending moment.

The usual formula for bending moment is of the form $\frac{\Delta \times L}{\text{bending momt.}} = f$, where $f$ is a factor varying from 20 upwards; the higher the factor the less is the intensity of the bending moment. In the above the scale for the bending moment is—

$$1 \text{ in.} = \frac{\Delta \times L}{200}.$$

Hence—

$$\frac{\text{number of inches in curve}}{200} = \frac{1}{f} = \text{intensity of bending moment.}$$

So that we may by this method speak of a six or a seven-inch bending moment as representing the intensity of this factor.

To illustrate the foregoing, the following figures are shown :—

| | |
|---|---|
| Light draught river paddle steamer ... ... ... | Fig. 15, Plate LXII. |
| 500 ft. cargo steamer ... ... ... ... ... | Fig. 16, Plate LXII. |
| 500 ft. Atlantic passenger steamer ... ... ... | Fig. 17, Plate LXIII. |
| Large armoured cruiser ... ... ... ... ... | Fig. 18, Plate LXIII. |

NOTE.—The diagrams illustrating this paper are reproduced one-half full size of working diagrams.

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[This paper was originally prepared for and read at the Paris International Congress of Naval Architects, July, 1900, but it has been slightly revised since then. The subject was considered by the Congress to be of sufficient importance to call the attention to it of the different foreign societies, and in consequence this matter is now brought to your notice. A series of diagrams showing standardised curves of stability has been prepared for forms varying in fineness from that of a racing yacht to the fullest type of cargo steamer, but this paper is already probably heavy enough for a midsummer meal, and the consideration of the effect of form upon standardised curves of stability may reasonably be left for a cooler and more serious season of the year. But the attention of members of this Institution is seriously called to this subject of standardising the results of ship calculation.]

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## DISCUSSION.

Professor J. H. Biles (Member of Council): With reference to this standardising question, it may be noted that the results of modern experiments can also be brought to a standard, and in that way we shall probably be able, when we get this tank, to remove the feelings of jealousy that make one person keep another from knowing the results of his model experiments, for, if they are all brought to a standardised result, all models will be in the same dimensions, and one will not be able to identify them with any particular ships.

Sir Nathaniel Barnaby, K.C.B. (Vice-President): My Lord and Gentlemen, I think this is a very fine piece of work. Our attention here in Glasgow has been called to it by our friends in Paris. We have gained a great deal of information from the Paris Congress, at which the subject of standardisation was considered in some papers read, one of them, I believe, by Professor Biles himself. We were asked to consider what could be done to bring ourselves into accord with each other. Professor Biles has been good enough to prepare this paper for our consideration, and there are two others to follow on similar lines, one on Screw Propellers, and the other by M. Hauser, of the French Navy, upon the "Adoption of a Rational System of Units in Questions of Naval Construction." I think Professor Biles has done very great service in putting before us in quite a simple manner the way in which we can be brought to act and work together in these calculations for the help of each other. While people in different countries are often separated by diplomatic methods, there is a great tendency to bring us together by what I may call co-operation in such an Association as this; and I have said to my friends sometimes that we feel, in looking at the work of another man, even though he belongs to another country, as much interested in it, and in getting it right, as if it were our own. In that way I think that the Germans, French, English, Russians, and the rest of us are being brought gradually to feel that we are one of a great company all working together for the common good, and, therefore, for the good of each nation.

Mr. F. K. Barnes (Vice-President): My Lord, I have nothing to say, except that I agree with every word that has fallen from Sir Nathaniel Barnaby, and I think it will be a source of very great good if this proposal is carried out.

Mr. J. Macfarlane Gray (Member of Council): My Lord and Gentlemen, the advantage of a system of generalisation must have frequently suggested itself to the minds of those much engaged in such work. To make a complete system of standardisation requires just that extensive knowledge of the whole subject which the author possesses in a pre-eminent degree. In my recent paper, "The Geometry of Engine-Balancing," I introduced a system of what I called *unification*, a minor application of the principles fully worked out by Mr. Biles at the Paris Congress, and in this paper. In this, I eliminated altogether the use of artificial units such as *foot*, *mètre*, *ton*, *kilogramme*, &c., from the expression for the result, and I think that Mr. Biles might perhaps have got rid of these also in the standardisation, had he tried.

Mr. Sydney Barnaby (Member of Council): My Lord and Gentlemen, there is so very much in this paper which is interesting to me, that I should like to have been able to study it more carefully. It treats in a very complete way, so far as I can judge, of various methods of standardising most of the calculations connected with ship design. Some of them we have probably been in the habit of employing ourselves, as, for example, the way of comparing the

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distribution of displacement by laying off the curves of sectional areas of the ships to be compared on a common base line to represent their length, and taking a common ordinate to represent the midship section; but Professor Biles has gone very much further than this, and he has shown that nearly all the calculations with which we are familiar may be dealt with in a somewhat similar way. He does not propose to hamper the designer by attempting to assimilate the forms of ships at all, but only to enable him to compare one form with another upon a uniform plan, and if it should be determined at some time to use a common unit of measurement—a metrical system of measurement, such as we are to have described to us in one of the following papers—it would be quite easy to transform the units which Professor Biles uses, into what we may call International Units. At the present time, the diagrams would be only useful for English and American shipbuilders, as the units chosen are feet and inches, but I think they will be found ultimately to be of wider value and of international interest.

Professor J. H. Biles: My Lord and Gentlemen, I would just like to say one word with reference to Mr. Barnaby's remarks about the results being only of use at present to those who use feet and inches. He probably did not notice that I suggested that those who use the metric system should set off their results, not on a basis of 10 inches, but on the basis of a quarter of a metre, which is 9·84 in. Then, although they would not be absolutely the same, yet to the eye they would be identical, and for a great many purposes of comparison, one diagram could be placed over the other; there would be a very small difference, which could be easily allowed for. I think, therefore, if we can get an agreement as to a system of standardising, we shall not have any difficulty in applying it. Another point is that we ought, before this subject is closed to-day, to take some action in order to consider what we shall do in furthering the intention that the International Congress had in referring this paper on to this Institution. I think, if no one else has any suggestion to make, that I would propose, if I may be allowed to, that this matter should be referred to the Council, to see whether we can form a small committee to consider it, for the purpose of sending it back to the International Congress, either now or at some future time, with a view to giving effect to what has been suggested in this paper.

Dr. Elgar: I shall be very pleased to move that.

Mr. Macfarlane Gray: I second that.

The President: Dr. Elgar has moved and Mr. Macfarlane Gray has seconded, that the question with which this paper is concerned be referred to the Council for consideration.

The motion, having been put to the meeting, was carried unanimously.

The President: Ladies and Gentlemen, as a layman I can only say that it appears to me that Professor Biles has dealt with one of the most difficult questions of the day in the most admirable manner, and I beg to move a vote of thanks to him for his paper.



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# ON SCREW PROPELLERS.*

Abstract of Two Papers by Mons. S. DRZEWIECKI.

[Read at the Summer Meetings of the Forty-second Session of the Institution of Naval Architects, June 27, 1901; the Right Hon. the Earl of GLASGOW, G.C.M.G., LL.D., President, in the Chair.]

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PROPELLERS have usually been designed up to the present time by means of practical rules deduced from trials and comparisons, or from empirical formulæ having no connection among themselves, depending chiefly on workshop tradition, and personal prejudices of the designer, varying in different countries, and showing generally very little knowledge of the working of screws in water. Imperfect as this method is, so long as screws of large diameter and slow rate of revolution were dealt with, giving moderate speeds, the approximate formulæ sufficed, and often gave remarkably efficient propellers, although sometimes resulting in surprises and failures. But now that higher speeds are sought, by increasing the rate of revolution of the engine, the screw conditions become more difficult, the old rules are no longer suitable, and others more scientific are required, based on the laws of fluid resistance, which will make it possible to calculate exactly all the elements to suit varying conditions of power, speed, and revolutions.

Some years ago the author presented to the Association Technique Maritime a paper on the general theory of propellers, based on the resistance of a submerged plane. Encouraged by its favourable reception, he endeavoured to obtain further proof of its correctness, by comparing the results of the trials of a large number of screws with the results predicted by his theory, and found them in close agreement. He has found it possible to explain hitherto obscure peculiarities in their action, and to discover the *rationale* of the empirical rules in use. To recapitulate briefly the principles upon which the theory is based, the author has shown that an element of a narrow inclined plane moving through a fluid experiences a resistance which can be resolved into two components, one of which represents useless resistance, and the other useful thrust, which is perpendicular to the first. These two component forces are related to each other in a manner independent of the speed, and depending only on the angle made by the plane to the direction of motion. He has shown that the relation between those components is a minimum when the plane is inclined at an angle of a little less than 3°.

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* The original of this paper, in French, appeared in the Bulletin de l'Association Technique Maritime (1900).

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In order to get the best effect from the blade of a propeller, each element of it should have this angle of incidence. In a propeller, the "angle of incidence" becomes the "slip angle." The surface fulfilling this condition is a helicoidal one of variable pitch. The author proceeds to show how to draw the variable pitch to produce the required result, and he calls a "normal screw" one in which this angle of incidence has been secured for every part of the blade, and in which the diameter of the boss bears a certain fixed ratio to the diameter of the screw. A screw of uniform pitch can be altered so as to give the desired constant angle of incidence, by twisting the blades in the boss to a greater pitch. The chosen angle of incidence, which is 2° 52′, corresponds to a mean pitch equal to 1·25 times the diameter, the latter being fixed by the author as the advance per revolution multiplied by $\frac{3}{\pi}$. He proposes to use certain definite shapes of blade, the curved outlines of which can be mathematically expressed. Propellers designed according to this theory will have all their dimensions, such as diameter, pitch, surface, diameter of boss, &c., expressible as a function of one of them. He proposes to state them in terms of the advance of the screw per revolution, and he thinks that great benefits would result if all propellers were designed upon some uniform theory, such as he sets forth.

### On the Choice of the Elements Determining Screw Propellers.

All screw propellers which give satisfactory results may be said to approach more or less perfectly to a common ideal type, of which the elements are strictly determinable by the general laws of mechanics and fluid resistance, and these are mutually related in a manner explained by the author in a paper communicated to the International Congress of Naval Architects in Paris in 1900.

The name of "normal screw" has been given to a perfect type of screw determined by exact calculation. It has been shown that all the geometric dimensions are proportional to one of the elements, the advance per revolution. It follows that, if all the elements of the "normal blade" be expressed in terms of the advance per revolution, a series of abstract numbers will be obtained, which will be the same for all "normal blades" without exception. Whatever may be the type of boat to which they belong, or the speed, number of revolutions, &c., the screws will differ only in size and in number of blades. These abstract numbers can be grouped in a single table, in such a manner that it will only be necessary to take as unity any given advance per revolution, when it will be possible to determine at once all the dimensions of the "normal blade" suitable for this rate of advance. The author recommends that experiments should be made with a large number of "normal screws" varying slightly from each other, in order to ascertain with certainty the proportions suitable for obtaining the best results.

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A table is given showing the proportions of the "normal blade." As this is designed to have a constant angle of incidence equal to 2° 52′, it follows that the pitch is different at different radii, being greatest at the extremity of the blade.

The number of blades to be employed is a function of the midship section, the advance per revolution and the hull efficiency.

The proposed method of designing screws will greatly simplify the drawing process, since one screw will differ from another only in scale. In small vessels, where twin screws cannot be used, and the engine requires to run faster than would be possible with one normal screw, the author recommends multiple screws placed one behind the other on the shaft.

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# ADOPTION OF A RATIONAL SYSTEM OF UNITS IN QUESTIONS OF NAVAL CONSTRUCTION.*

By M. HAUSER, Chief Engineer in the Navy (retired).

[Read at the Summer Meetings of the Forty-second Session of the Institution of Naval Architects, June 27, 1901; the Right Hon. the Earl of GLASGOW, G.C.M.G., LL.D., President, in the Chair.]

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WHEN the Electrical Congress met in 1881, the situation was almost identical with that in which we are now placed. Although the British Association, as early as 1860, had taken the initiative in the unification of electrical measurements, the members of that Congress had no common technical language, consequently the results which their isolated experience furnished could only be compared with difficulty. Once the Congress had adopted the method of calculation known as the C.G.S. system, now universally employed, the results were comparable, and electricity took the forward step, which has been the most salient characteristic of industrial and scientific life during the last twenty years of the past century. We have not made the same advance. Whenever we wish to compare materials, speed, strength, work, or power, we instantly meet obstacles, which I can only compare to the Custom-house barrier, which so jealously surrounds most nations; we pay heavy taxes under the form of unproductive work and precious time lost.

It is needless to remind those who have been brought up to use the decimal and metric system, how often they have cursed the many subdivisions of the inch. We have endless trouble in recognising the 7ths, 8ths, and 12ths of an inch, and it is very difficult for us to understand, at first sight, that an iron plate $\frac{5}{8}$ of an inch is about 16 mm. thick. I might add that it is also not easy to form an immediate idea of thickness when it is defined by the weight of a superficial unit; it is not without a little calculation that we realise that a plate weighing 15 lbs. per square foot is about 9·4 mm. thick.

As to speeds, we do not always know if it is a question of the nautical mile of 1,852 m. or of the statute mile of 1,609 m. (1,760 yards).

Regarding the unit of work, we find sometimes the kilogrammètre, sometimes the foot-pound (0·138 kilogrammètre), sometimes the foot-ton (309·89 kilogrammètres).

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* The original of this paper, in French, appeared in the Bulletin de l'Association Technique Maritime (1900).

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As to power, we hesitate, according to circumstances, between the *cheval-vapeur* of 75 kilogrammètres and the horse-power of 76·04 kilogrammètres a second (33,000 pounds × 1 ft. a minute).

If it is a question of the displacement of a ship, we have to choose between the *tonne* of 1,000 kg. and the ton of 1,016·048 kg.

Finally, if it is a question of pressure, or of tension per unit of surface, we reckon in—

Kilogrammes per square centimètre.
Atmospheres (1·033 kg. per square centimètre).
Kilogrammes per square millimètre.
Pound per square inch (·0703 kg. per square centimètre).
Ton per square foot (1·0937 kg. per square centimètre).

The physicists are more fortunate. They have to consider densities, co-efficients of expansion, &c., upon which nationality has no influence, as they are merely ratios.

Chemists also are happier than we: the definite proportions of their combinations are independent of local weights and measures.

Electricians are more fortunate than we, for they have had the courage to sponge the slate, and have created a uniform language of their own.

For ourselves, we remain the slaves of the table of conversion, the aid to memory, and the slide rule. Is it wise to submit to such restraints?

I shall not speak of geometrical units—length, surface, volume, weight. Adopted for so long in many countries, and tolerated or made optional in others, the metric system progresses little by little, and it is very probable that its undoubted usefulness, rendered more and more evident by the increasing frequency of international communications, will lead to its universal adoption, thus relegating to the past, even as in France, those inconvenient and superannuated methods, which it ought to displace.

It is to three values in constant use in our work, that I wish to call the attention of the Congress. Would it not be advantageous to connect to the C.G.S. system, employed already by an important class of engineers and manufacturers, the practical units of work, power, and pressure per unit of surface? Our unit of work is the kilogrammètre, that of the Anglo-Saxon countries is the foot-pound (0·138 kgm.), or the foot-ton (309·89 kgm.); lastly, that of the C.G.S. system is the erg ($1{\cdot}019 \times 10^{-8}$

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kgm.).* But, as the employment of the erg in current applications gives generally too high a number, electricians have adopted, as practical unit, the joule, equal to $10^7 \times$ erg, or $1{\cdot}019 \times 10^{-1}$ kgm., or 0·1019 kgm. We propose to reckon work in kilojoules, or 101·9 kgm. (more exactly, 101·937).

Our unit of power is the *cheval-vapeur* of 75 kilogrammètres, that of Anglo-Saxon countries the horse-power of 76·02 kilogrammètres. We should propose to adopt as unit of power, the kilowatt, generally employed by electricians, and which, corresponding to the kilojoule, would express by the same number the work done per second and the power developed by the same apparatus. One might say that an engine furnishing 1 kilojoule per second is an engine of 1 kilowatt. We should get rid of the coefficient $\frac{1}{75}$, with which we overload our calculations.

In this system of units, the work would be expressed by numbers differing but little from the numbers of actual kilogrammètres divided by 100 (exactly $\frac{1}{101{\cdot}937}$), and the power by numbers which would be equal to the actual numbers multiplied by—

0·73575 for *chevaux-vapeur.*
0·74594 for horse-power.

or divided by—

1·359157 for *chevaux-vapeur.*
1·340588 for horse-power.

The joule and the watt, as well as their derivatives, are units of measure of energy, and of energy per unit of time, whatever form it takes, mechanical, calorific, or electrical. It is incorrect to consider them as belonging exclusively to electrical science. It is, besides, more rational to substitute them for kilogrammètres and horse-power respectively, as, in numbers of cases, electrical energy is transformed into mechanical energy, or conversely; and it is easier, with the same unit, to follow and value a like energy under all its forms. Thus, when a motor drives a dynamo, is it not easier to value with the same unit the power furnished by the motor, and that returned by the instrument yielding the current? and is it not easier at once to see that, the power indicated by the motor being 500 kilowatts, and that furnished by the dynamo 400 kilowatts, the efficiency of the whole is 0·80, instead of having to convert either of the units?

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* In the C.G.S. system, the unit of force, or dyne, is the force which, acting upon the unit of mass, communicates to it in the unit of time the unit of acceleration. The unit of mass being the mass of a gramme, and the unit of acceleration the centimètre, it follows that the dyne is equal to

$$\frac{1 \text{ gr.}}{981} = \frac{1 \text{ kg.}}{981} \times 10^{-3} = 1{\cdot}019 \text{ kg.} \times 10^{-6}.$$

The work of a dyne displacing its point of application of the unit of length (centimètre) or erg, is, therefore, equal to $1{\cdot}019 \text{ kg.} \times 10^{-6} \times 1 \text{ m.} \times 10^{-2} = 1{\cdot}019 \times 10^{-8}$ kilogrammètres.

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Further, two dynamos are often coupled, the one working electrically, the other mechanically, and it would be hardly rational to express in kilowatts the electrical power furnished by the generator, and in *chevaux-vapeur* the mechanical power which the receiving dynamo gives.

Moreover, it appears to us indispensable, if it were only for the sake of simplifying technical language, to adopt a practical unit of pressure, or tension reduced to the unit of surface. The C.G.S. unit would be the dyne square centimètre.

In order to conform to precedents, and to immortalise the name of the great physicist, whose researches upon the pressures of steam are the basis of so many calculations, we propose to give the practical unit the name of Regnault. In order to establish the connection between this unit and that of work, we will define it in the following manner:—

"The *kilo-regnault* is the load per unit of surface, which, applied to 1 square mètre, and displacing it through 1 mètre, gives 1 kilojoule of work."

The result is that the *kilo-regnault* is a load of 101·937 kg. per square mètre. The *regnault* will be, therefore, equal to 1·019 kg. $\times$ $10^{-1}$ per square mètre, or 1·019 kg. $\times$ $10^{-5}$ per square centimètre; and, as the dyne is equal to 1·019 kg. $\times$ $10^{-6}$, the *regnault* is a load of 10 dynes per square centimètre.

It may be objected, perhaps, that the adoption of this unit leads, in ordinary work, to an excessive number of figures; but this objection is only apparent, for it is simpler to express the same pressure by 1,006 *kilo-regnaults*, than to say that it is 10·25 kg. per square centimètre. The kilogramme must ordinarily be subdivided, whilst the *kilo-regnault* need not be. Similarly it is easier to say that the breaking load of steel is 739 *mega-regnaults*, than to say that it is 75·35 kg. per square millimètre.

It may be remarked that the *regnault* plays, in mechanics, the same part that the volt plays in electricity. It measures the pressure, or difference of potential, of steam, the product of which, by delivery in unit of time, is a measure of power, just as, in an electrical apparatus, the product of the electromotive force (expressed in volts) by the quantity delivered in the unit of time $\left(\frac{\text{coulombs}}{t}, \text{ or ampères}\right)$ measures the power per second (watts).

The Tables of Conversion annexed to this paper will facilitate the change from a number of units of one system into a corresponding number of another system.

In summing up, we would propose that an international understanding should be established for the adoption of practical units derived from the C.G.S. system for work, power, and pressure or tension per unit of surface. It is for the scientific and

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technical world at large to inquire in what manner this adoption is possible, and the object of this paper would be fully attained, if it gave rise to an improvement in our technical terminology.

TABLE OF CONVERSION OF WORK.

| Kilojoules. | Kilogrammètres. | Kilogrammètres. | Kilojoules. |
|---|---|---|---|
| 1 | 101·937 | 1 | 0·00981 |
| 2 | 203·873 | 2 | 0·01962 |
| 3 | 305·810 | 3 | 0·02943 |
| 4 | 407·747 | 4 | 0·03924 |
| 5 | 509·684 | 5 | 0·04905 |
| 6 | 611·621 | 6 | 0·05886 |
| 7 | 713·558 | 7 | 0·06867 |
| 8 | 815·494 | 8 | 0·07848 |
| 9 | 917·431 | 9 | 0·08829 |

TABLE OF CONVERSION OF POWER.

| Kilowatts. | Chevaux-vapeur. | Chev. vap. | Kilowatts. | Kilowatts. | Horse-power. | Horse-power. | Kilowatts. |
|---|---|---|---|---|---|---|---|
| 1 | 1·359157 | 1 | 0·73575 | 1 | 1·340588 | 1 | 0·745942 |
| 2 | 2·718315 | 2 | 1·47150 | 2 | 2·681175 | 2 | 1·491883 |
| 3 | 4·077472 | 3 | 2·20725 | 3 | 4·021763 | 3 | 2·237825 |
| 4 | 5·436629 | 4 | 2·94300 | 4 | 5·362350 | 4 | 2·983766 |
| 5 | 6·795787 | 5 | 3·67875 | 5 | 6·702938 | 5 | 3·729708 |
| 6 | 8·154944 | 6 | 4·41450 | 6 | 8·043525 | 6 | 4 475649 |
| 7 | 9·514101 | 7 | 5·15025 | 7 | 9·384113 | 7 | 5 221591 |
| 8 | 10·873259 | 8 | 5·88600 | 8 | 10·724701 | 8 | 5·967533 |
| 9 | 12·232416 | 9 | 6·62175 | 9 | 12·065288 | 9 | 6·713474 |



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TABLE OF CONVERSION OF PRESSURE PER UNIT OF SURFACE.

| Kg. per sq. cm. | Kiloregnaults. | Kiloregn. | Kg. per sq. cm. | Kg. per sq. mm. | Mégaregnaults. | Mégaregn. | Kg. per sq. mm. |
|---|---|---|---|---|---|---|---|
| 1 | 98·1 | 1 | 0·01019368 | 1 | 9·81 | 1 | 0·1019368 |
| 2 | 196·2 | 2 | 0·02038736 | 2 | 19·62 | 2 | 0·2038736 |
| 3 | 294·3 | 3 | 0·03058104 | 3 | 29·43 | 3 | 0·3058104 |
| 4 | 392·4 | 4 | 0·04077472 | 4 | 39·24 | 4 | 0·4077472 |
| 5 | 490·5 | 5 | 0·05096840 | 5 | 49·05 | 5 | 0·5096840 |
| 6 | 588·6 | 6 | 0·06116208 | 6 | 58·86 | 6 | 0·6116208 |
| 7 | 686·7 | 7 | 0·07135576 | 7 | 68·67 | 7 | 8·7135576 |
| 8 | 784 8 | 8 | 0·08154944 | 8 | 78·48 | 8 | 0·8154944 |
| 9 | 882·9 | 9 | 0·09174312 | 9 | 88·29 | 9 | 0·9174312 |
| **Lbs. per sq. in.** | **Kiloregnault.** | **Kiloregn.** | **Lbs. per sq. in.** | **Tons per sq. ft.** | **Kiloregnault.** | **Kiloregn.** | **Tons per sq. ft.** |
| 1 | 6·897367 | 1 | 0·14498 | 1 | 107·292413 | 1 | 0·009320 |
| 2 | 13·794734 | 2 | 0·28997 | 2 | 214·584827 | 2 | 0·018640 |
| 3 | 20·692101 | 3 | 0·43495 | 3 | 321·877240 | 3 | 0·027961 |
| 4 | 27·589467 | 4 | 0·57993 | 4 | 429·169654 | 4 | 0·037281 |
| 5 | 34·486834 | 5 | 0·72491 | 5 | 536·462067 | 5 | 0·046602 |
| 6 | 41·384201 | 6 | 0·86990 | 6 | 643·754480 | 6 | 0·055922 |
| 7 | 48·281568 | 7 | 1·01488 | 7 | 751·046894 | 7 | 0·065242 |
| 8 | 55·178935 | 8 | 1·15986 | 8 | 858·339307 | 8 | 0·074563 |
| 9 | 62·076302 | 9 | 1·30485 | 9 | 965·631721 | 9 | 0·083883 |

Power indicated by a steam engine cylinder—

In *chevaux-vapeur*, $F = \frac{D^2 C N P}{0{\cdot}28648}$; in kilowatts, $\Phi = \frac{D^2 C N R}{38{\cdot}1977}$,

where D = Diameter of cylinder in mètres.
C = Length of stroke in mètres.
N = Revolutions per minute.
P = Pressure in kilogrammes per square centimetre.
R = Pressure in regnaults.

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## DISCUSSION.

Mr. Sydney W. Barnaby (Member of Council): My Lord and Gentlemen, I should like to say a few words on Mons. Drzewiecki's proposals. These two papers of Mons. Drzewiecki's contain two propositions. The first proposition is that there is one particular form of screw, having a mean pitch equal to about 1·25 times the diameter, and having its pitch disposed in a particular way, which will give better results than any other form of screw. The second is that, assuming that to be true, it would be desirable to use this form and no other, so that one drawing of a screw would suffice for all purposes, the only variation being in the number of blades, and in the size and number of the propellers to suit different cases. Now, as regards the first proposition, the author has not mentioned Mr. Froude's name; but I am sure he is aware that his contention is not in accordance with Mr. Froude's experimental results. Mr. Froude has expressed his surprise that his experiments did not show any superiority of one pitch ratio over another, but, on the contrary, there appeared to be little difference in the efficiency of screws differing very widely in pitch ratio. Before the date of these experiments, there was, I think, some consensus of opinion that pitch ratios of about the value of 1·25 (which the author now proposes to use) gave the best results. But as this does not seem to be established, it seems to me a pity to limit ourselves too much in the design of screws, in order to obtain uniformity of pattern; for, after all, the real difficulty is not in the drafting of the propeller, but in the correct estimation of the speed of the following current, the propulsive co-efficient of the vessel, and the effect of the screw upon the resistance of the vessel. The author's proposals would not help us in any of those difficulties, which would still remain. But the effect of the proposed limitation to a single model would be that, for a given horse-power and speed, the revolutions would be pretty definitely fixed, because it is well known that maximum efficiency can only be attained with a given pitch ratio over a rather limited range of slip ratio. A variation from the proper rate of revolution for a given power and speed could only be made, in such a case, by some sacrifice of efficiency, or by multiplying the number of screws used. While, therefore, I think that the Institution is indebted to the author for his attempt to simplify some of our calculations and render comparisons more easy, I am of opinion that, if the proposals were adopted, they would not help to overcome any real difficulties, and they would impose unnecessary and undesirable restrictions upon the freedom of design.

Mr. J. Macfarlane Gray (Member of Council): I agree with every word that Mr. Barnaby has just said about this paper. In the year 1856, I started to make a similar standardisation of the screw propeller surface; I completed the calculations, and only when I at length applied the results to practice, I found what, until then, I was ignorant of, namely, that the uncalculable motion of the wake water vitiated all my calculations. I bundled up the papers and never looked at them again, until, on my retirement a few years ago, I found amongst my things a dusty package labelled, "Screw Dreams," reminding me of my former work and the name I had given to the castaway. Mr. Barnaby has detected the same omission in the paper now read.

Mr. Hauser's paper says: "It appears to us indispensable, if it were only for the sake of simplifying technical language. . . ." By all means *simplify;* but the paper entirely fails to do so. The monuments to scientists ought to stand in their works or in our public parks, and not upon the tips of our tongues, when we speak of the science they taught us. We must go back to our remote

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ancestors for a practical example of how to name units. They gave us generally monosyllabic, pithy, clear names—*yard, inch, pound, mile:* always a short word to denote a frequent idea. How do these compare with *megaregnault, kilojoule, kiloregnault?* The opposite of simplification has been perfected in the use of *Joule* with a capital J for 41,810,000 (or thereabout) ergs, and *joule* with a little j for 10,000,000 ergs.

Sir Nathaniel Barnaby, K.C.B. (Vice-President): My Lord and Gentlemen, I would like to propose that the thanks of the Institution be given to the authors of these two papers, and that they be referred, like the former ones, to the Council for consideration.

Mr. Archibald Denny (Member of Council): I beg to second what Sir Nathaniel Barnaby proposes. Mons. Hauser has my entire sympathy, and, if we could only get our people here to adopt the metric system in its entirety, we would save ourselves a great deal of trouble. It is officially within the knowledge of this Institution, and I would like to inform those who do not know it, that I am (as representing our Institution) at present engaged on a representative committee which is endeavouring to standardise steel sections. That committee has already taken some evidence, but I do not know how long it will be before we make our report. The bulk of the papers which have been read this time have dealt more or less with standardisation. I think standardisation is in the air, and anything that this Institution can do to help forward that good work, will be well done. I beg to second what Sir Nathaniel Barnaby has proposed.

Professor J. H. Biles (Member of Council): My Lord and Gentlemen, just to prevent any misapprehension as to the use of the word "standardisation," in relation to what Mr. Denny has said, I should like to remark that standardising, in the case of the committee he is dealing with, is perhaps better expressed as being an attempt to make things uniform. What I had in view, however, and what I think some of the authors of the other papers had in view, is what is expressed by Mr. Macfarlane Gray as *unification*—bringing things to a common unit—so that while, as Mr. Denny says, we have standardisation in the air, the two kinds of standardisation are on totally different lines, and must not be confused in any way. The standardising that I suggest is one absolutely independent of dimensions altogether. The paper read by Mons. Hauser is an attempt to make a name for physical units, and has no relation whatever to standardising results of geometrical forms. They are totally different questions. I quite approve of the objects, if I may be allowed to say so, that the writer of that paper has, in attempting to make a uniform system of units that shall be independent of nationality. The question of whether we should say a *Kilo-regnault* or a *squeeze*, or some other name, could very well be left to the Council. I think the subject of the paper on screw propellers that we have had to-day hardly comes within the scope of the question of standardising, except as a method of giving two propellers of the same form different dimensions. The question as to whether we could make all propellers of the same form is quite apart from that of standardising, and would be analogous to attempting to adopt a uniform shape of ship.

The President: I understand, Professor Biles, you approve of the proposal to refer these papers also to the Council?

Professor J. H. Biles: Certainly.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): It has been moved and seconded that these two papers by Mons. Drzewiecki and Mons. Hauser be referred to the Council.

The motion, having been put to the meeting, was carried unanimously.

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Mons. S. Drzewiecki (Associate): My Lord and Gentlemen, I do not speak English, and I can only express my gratitude for the honour I have received in the selection of my paper by your Institution.

Mr. Thomas (Visitor): My Lord, the only word I could add to the discussion of this question is that it is thoroughly appreciated in America by our engineers. Together with the other requirements of speed and efficiency, the standardisation of units is a very necessary consideration. One point which has not been brought out here, but which has been agitated in America, is the standardisation of machine parts. Certain units are used in various ships, and are designed in practically the same way, as, for instance, both for circulating pump engines and for a pump of a certain power. There is no reason why we should not have a great many units made from exactly the same drawings, so that they would be more or less interchangeable, and when one set has been made, the next set could be made more cheaply.

The President (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.): Ladies and Gentlemen, I beg to move a hearty vote of thanks to Mons. Drzewiecki for his very interesting paper, and I will also ask you to approve of our Secretary sending to Mons. Hauser the thanks of the Institution for the paper which has just been read.

### Written Communications.

Mons. A. Hauser: My Lord and Gentlemen, allow me to tender my thanks to the Institution of Naval Architects for so kindly accepting the paper which I had presented to the Congress of Naval Architects and Construction in 1900. I should like to add a few remarks, which the year that has elasped since then has suggested to me. At a time when the increasing facilities of communication bring different nations daily into closer contact, every hindrance to their intercourse, either commercial or intellectual, becomes a drawback, more especially in the realm of intellect. Every nation that wishes to keep ahead in the struggle, must follow the progress made by its neighbours in every line of knowledge, and the only means by which this can be done, is by a right understanding of its neighbour's tongue. If, so far, no universal language has been possible, let us at least make every use of that international speech which we do possess, namely, mathematics. For, here at least, all nations are on a common footing; all use the same methods of calculation, whether arithmetical, algebraical, or analytical. But, as though to neutralise all the advantages that might be derived from this fact, they proceed, although following identical lines of thought, to start with different data, and thus arrive at dissimilar results; as, for instance, in the variety of units employed in formulæ and equations. Thus, all the advantage of a common scientific language is lost in the application of the methods. If the units of work, power, and load per unit of area, be connected to the C.G.S. system, experimental results can then be compared with ease: no doubt the comparing of observed facts is a powerful factor of progress in naval architecture, as in all experimental science. But we must consider, besides, all the simplifications which result from this, in calculations and researches, and the gain in time and work, which all active nations will appreciate. In consenting to discuss this subject, the Institution of Naval Architects is giving it great assistance, and, perhaps, after due consideration and discussion, not only its co-operation, but that of the British Admiralty, may be obtained.

Mr. J. M. Adam (Associate): As a small contribution to the discussion of a rational system of units, a relationship may be pointed out between the metric and the British standards of length, through

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the value of $\pi$, viz., 1 in. : 80 mm. : : $d : \pi$. For example, in constructing a propeller pitch diagram, the base may be pricked off in spaces of 20 millimètres. Let these spaces each represent 6 in. radius, or 1 ft. diameter. Then a quarter-inch scale used for the staff will represent the pitch in feet within the limits of graphic error:—

$$1 \text{ in.} \times \pi = 3{\cdot}1416 \text{ in.}$$
$$80 \text{ mm.} \quad = 3{\cdot}1496 \text{ in.}$$

a discrepancy of about $\frac{1}{400}$th of the length.

With reference to M. Drzewiecki's paper on screw propellers, the writer desires to concur in the propositions stated in the introductory paragraph of the abstract, and in the aspirations of the remainder of the first paper for the adoption of a varying pitch to follow the successive acceleration of the current, and also to determine and standardise a best possible set of proportions; but he demurs from the assumption that the resistance of a revolving screw can be determined by the resistance of a submerged plane, as the component forces brought into play are, in the former case, complicated. The component of resistance to an inclined plane at right angles to the useful thrust, and described as representing *useless resistance*, becomes, in a rotating vane, tangential; whereas, to give quite analogous results, it must be radial, with a centripetal reaction. The form proposed, or any helicoidal form, fails to meet this requirement; which can be met only by a properly designed conoidal surface. This required centripetal reaction is provided in the geometrical form of the inner surface of a bird's wing, which appears to be the nearest, or perhaps the only analogue in nature to the instrument under review. Exception must also be taken to the proposition that the vane proposed, or any vane, can be uniformly altered in pitch by rotating it on the boss, whatsoever line be chosen for axis.

Monsieur S. Drzewiecki (Associate): My Lord and Gentlemen, as I am not proficient enough in the English language to take part in the discussion, allow me to add these few lines to what has been said:—The International Congress of Naval Architects, held in Paris in 1900, having, among other things, expressed a hope for the unification of measurement of the elements of the screw propeller, to facilitate comparison, I have tried to give a practical method for arriving at this unification in the paper whose abstract, unfortunately a little too short, has been submitted to you. In the theory of screw propellers which I laid before the Congress, and of which Mr. Sydney Barnaby has kindly made an abridged extract, embodying some of its principal points, I have only sought for a scientific method of rationally bringing together observed phenomena; and, when I proposed a special type of screw propeller, which I called the *normal* screw, it was not so much to consider it as the *perfect* propeller, best in all circumstances, as to offer it as a *standard* of comparison, having already proved that every blade of a propeller can always be replaced by a *normal* blade whose elements can be exactly calculated. Furthermore, in proposing to adopt the method for determination, in abstract numbers, of all the elements of screw propeller blades, which is a direct result of my theory, I have given the practical means for the unification of the measurements of these elements. This is the real object of the paper which I have had the honour of submitting to you.

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## CONCLUDING PROCEEDINGS.

The PRESIDENT (the Right Hon. the Earl of Glasgow, G.C.M.G., LL.D.) : Ladies and Gentlemen, the papers we have just heard read conclude the proceedings of the Institution as regards business at this Summer Meeting. It is for you, who have been at other meetings of the same sort, to say whether you consider this has gone off as satisfactorily as usual. We have heard Sir Nathaniel Barnaby say that he thinks this meeting will be memorable, if only for the two incidents to which he alluded, and which are to be referred to the Council for further consideration. I need not dwell upon them because they are still fresh in your minds. For my own part, I feel very proud at having presided on this occasion, and I hope the proceedings have been conducted to your satisfaction. Before we part there are several votes of thanks which I am sure you will wish to accord on this occasion. I think I had better take them all together. In the first place, I beg to move a vote of thanks to the Hon. the Lord Provost and the Members of the Corporation of Glasgow, for the reception they gave us at the Municipal Buildings. I am sure it must have been a pleasure to those of you who had never before been there, to see the magnificent way in which the Corporation of Glasgow have housed themselves, and I think we in Scotland have every reason to be proud of those buildings, and I observe that they do command the admiration of all those who see them for the first time. I will also ask you, Gentlemen, to concur in a vote of thanks to the General Reception Committee, of which the Lord Provost is chairman, and to Mr. George Strachan, the honorary secretary, for all the trouble they have taken. Also to Principal Story and the Senate of Glasgow University—we must do it before it happens—for the reception they are going to give us this evening, and for the other courtesies they have shown us on this occasion ; and, finally, to the owners of the various works which have been open to our inspection. Gentlemen, I beg formally to move a vote of thanks to these bodies for their courtesy and kindness to us.

(The motion was carried by acclamation.)

I think before we part, I should refer to Herr Bussley's invitation, which he made last night at dinner, to the Institution individually and collectively. He asked them to meet the Schiffbau Technische Gesellschaft at Dusseldorf next year, at a date which I think is not yet fixed, and he gave that invitation in a most cordial manner on the part of those whom he represented. I am sure all here present are most grateful to Herr Bussley, and to his brother members, for that invitation, but I think it will be in accordance with the usual practice if we refer it to the Council for consideration, and for further publication among the members of the Institution. Gentlemen, I understand it is your wish that that course be adopted, and that the Council should give a reply to Herr Bussley for his kind invitation.

Ladies and Gentlemen, I have been speaking of our business arrangements. As far as the arrangements for the amusement of the Institution are concerned, they have very nearly come to an end. I need hardly remind you, and the subject is nearest my heart, that this afternoon I shall be proud to welcome you all at Kelburne.

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Admiral the Hon. Sir E. R. Fremantle, G.C.B., C.M.G. (Associate): My Lord Glasgow, Ladies and Gentlemen, I have a very pleasing duty to perform, and that is, to propose a vote of thanks to Lord Glasgow, for the courtesy, consideration, and good judgment which he has shown, both in the chair and during the whole period we have been here. I think we Naval Architects are especially fortunate in having as our President this year one who, living close to Glasgow, is so thoroughly appreciated there. Last night Lord Glasgow modestly said that he did not know what qualification he had for being President of this Institution. He stated that he had been brought up as a Naval officer, and had consequently not had the opportunities of studying those problems of naval science which are so well debated in this Institution. But he did say that he hoped he possessed some of the qualifications of a good chairman; at least, he supposed that was the reason he had been asked to undertake those duties. I can assure him he was not mistaken, for we all appreciate his good qualities in that respect, and, as I stated before, we are very fortunate in having this year, for the visit of the Institution to Glasgow, one who represents not only in name, but also in character, all the best qualities which are known to be the birthright of Scotsmen. Without any further preface, I beg to propose a vote of thanks, and a very cordial one, to our Chairman, not only for the way in which he has filled the chair, but for all the arrangements he has made, with the municipal and other authorities of this great city, for our reception. I would further mention—although Lord Glasgow has, I think, already expressed it on our behalf—the satisfaction we feel at the presence here of so many representatives of other nations—French, Germans, Americans, Russians, and others who have been good enough to come, and who have aided the Institution so much by their works, towards the solution of those problems which have been brought before us. Gentlemen, I beg to propose a vote of thanks to Lord Glasgow, our President.

Mr. Francis Elgar, LL.D., F.R.S. (Vice-President): My Lord, Ladies, and Gentlemen, in rising to second the resolution that has been so well proposed by Admiral Fremantle, I have really nothing to add to what he said. The Institution of Naval Architects has always been exceptionally fortunate in its President, and we have never been more so than at present. The members of the Institution have had ample opportunities during these meetings of seeing for themselves the devotion of Lord Glasgow to the interests of the Institution, and of witnessing the very able manner in which his Lordship fills the Chair. We Naval Architects are proud to serve under Lord Glasgow as our President, and we feel it a great privilege and pleasure to work with him, as we are glad to do in every possible way, to promote the interests of the Institution. I have very much pleasure in seconding the resolution.

The resolution was carried by acclamation.

The President: Ladies and Gentlemen, I took the opportunity last night of saying that I thought it would be a very good rule if somebody else had always to return thanks for a man when his health is being drunk, or a vote of thanks to him is being proposed. But as I have to return my own thanks, I must say that I feel very grateful to Sir Edmund Fremantle and to Dr. Elgar for the kind manner in which they have proposed and seconded the vote of thanks which you have just given to me. I can only say it will be my endeavour to do my best to fulfil the duties of President of your Institution, and if I gain your approbation, it will be the greatest reward I can possibly receive.

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# OBITUARY NOTICES.

## THE RIGHT HON. LORD ARMSTRONG, C.B., D.C.L., F.R.S.

By the death of Lord Armstrong, the Institution has lost one of its most distinguished Vice-Presidents. The Right Hon. William George, First Baron Armstrong, C.B., D.C.L., F.R.S., died on December 27, 1900, at his residence, Cragside, Rothbury, Northumberland. He was born on November 26, 1810, and was the son of Mr. W. Armstrong, a merchant of Newcastle-on-Tyne, his mother being the daughter of Mr. William Potter, of Walbottle Hall, Northumberland. He was educated for the legal profession, and commenced his career in a solicitor's office. Later he studied law under his brother-in-law, who afterwards became Baron Watson. The study, as an amusement, of mechanical problems led to his finding his true vocation; and in 1847 he joined the firm of Donkin, Cruddas, Potter & Lambert, who owned engineering works at Elswick-on-Tyne. Hydraulic engineering had been the chief object of his researches, and, owing to his labours, the firm he had joined soon gained recognition as one of the chief representatives of this branch of mechanical industry.

The events of the Crimean War led him to devote his thoughts to the improvement of ordnance, and in 1858 a department for the manufacture of heavy guns was added to the Elswick Works. Lord Armstrong's experiments and researches culminated in the production of a design which quite revolutionised the art of constructing both naval and field artillery. The Armstrong system of gun construction was founded on the principles laid down by Professor Barlow, in which the need for initial tension in the exterior parts of the weapon was mathematically demonstrated. Lord Armstrong, then Mr. Armstrong, attained the desired end by shrinking hoops round an interior tube, in the way now so well known. Lord Armstrong early turned his attention to the improvement of ships' armour. He at first preferred wrought iron to steel; but later made experiments with a view to getting over the objectionable brittleness of the latter material by a process of toughening by immersion in oil.



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The firm of Armstrong was from the first closely associated with Naval matters through the supply of ordnance and hydraulic machinery. By the incorporation of a shipbuilding yard at Walker-on-Tyne, and by the establishment of the Elswick shipyard, the company became direct builders of both Naval and Mercantile vessels, many famous vessels being launched from these yards. Lord Armstrong held pronounced opinions on warship design. He was opposed to the construction of very large battleships, being greatly in favour of small, heavily armed vessels, of which the *Staunch* class were an example, and he held that armament should be considered before armour. In 1889, during the Newcastle Meeting of the Institution, he said: "We may come to this conclusion, that, if we render a ship absolutely safe from being sunk by modern artillery, we shall simply eliminate its power of sinking anything else. . . . What I have learned on this occasion only tends to increase the disfavour with which I look on the armoured class of battleships as a whole. . . . We cannot have a numerous Navy, if we devote our resources chiefly to the production of such ships. We require for the protection of our commerce and for general services that our Navy should be extremely numerous. The only means of obtaining a Navy of sufficient numbers for that purpose is to restrict our expenditure upon ships of this class, and increase our expenditure upon ships of the cruiser class."

Lord Armstrong was much interested in all branches of physical research. He constructed a hydro-electric machine by which electricity was generated through steam escaping at high pressure from an insulated boiler. A remarkable electrical experiment which he made, more than fifty-three years before his death, was described in a memoir he wrote in 1897, shortly after he had again taken up the subject, with an interval of half a century intervening.

Lord Armstrong was a Fellow of the Royal Society, a Past President of the Institution of Civil Engineers, and a Past President of the Institution of Mechanical Engineers. He was elected a Vice-President of this Institution in 1882, and contributed to the Transactions a paper on "The Application of Hydraulic Power to Naval Gunnery" in 1888.

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## ADMIRAL OF THE FLEET SIR JOHN EDMUND COMMERELL.

Admiral of the Fleet Sir John Edmund Commerell, G.C.B., V.C., who died at his residence in Rutland Gate on May 21, 1901, was an Associate of the Institution, having been elected in the year 1894. He was the second son of Mr. J. W. Commerell, of Strood Park, Horsham, and was born in London, January 13, 1829. He entered the Royal Navy in 1842, and saw active service during the first China war. In 1848 he was promoted to the rank of Lieutenant and served in the Mediterranean, in the Channel, and in the Baltic. On September 29, 1855, he was given the command of a gunboat, which he took from the Black Sea into the Sea of Azoff, and it was during this trip that his distinguished services won him the Victoria Cross. He was back in Chinese waters in 1859, when he was Commander of the *Fury*, and on June 25 he led a division of seamen who landed to attack the Taku Forts. In 1866 he was in command of the *Terrible* in assisting to lay the Atlantic cable, for which service he was created a C.B. In 1871 he was Commander-in-Chief on the West Coast of Africa and Cape of Good Hope Station. Whilst ascending the river Prah, when operating against the Ashantis, he was shot in the lungs and was invalided home. In 1874 he was nominated K.C.B.; in 1876 he became a Rear-Admiral; in 1877 he was second in command in the Mediterranean; in 1881 he became a Vice-Admiral; in 1886 he was promoted to Admiral; and in 1892 became an Admiral of the Fleet. In 1882 he was Commander-in-Chief on the North American Station, and appointed Commander-in-Chief at Portsmouth, and was made G.C.B. in 1887. He was twice elected as member for Southampton, but resigned on being appointed Commander-in-Chief at Portsmouth.

Sir Edmund Commerell took great interest in engineering matters and mechanical appliances. He invented a jury rudder, and was instrumental in introducing certain new copper alloys, being connected with the Phosphor Bronze Company in its earlier days.

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## SIR RAYLTON DIXON.

The death of Sir Raylton Dixon, which occurred on July 28, 1901, after a long and painful illness, has deprived the Council of the Institution of one of its most representative and popular members.

Born at Newcastle-on-Tyne in 1838, Raylton Dixon, son of Mr. Jeremiah Dixon, of Ballawray, Ambleside, served his apprenticeship with Messrs. Coultas & Parkinson, and afterwards with Messrs. Charles Mitchell & Co., of Newcastle-on-Tyne. At the early age of twenty-one, his remarkable technical ability and business capacity having already asserted themselves, he was appointed manager of the branch shipyard of Messrs. Richardson, Duck & Co., of Middlesbrough. In 1863 that firm dissolved, and Mr. Dixon took over the business in partnership, first with Mr. Backhouse, and, at the latter's retirement, with his brother, Mr. Waynman Dixon.

Sir Raylton Dixon's career was one of great success. Gifted with remarkable energy and a genius for organisation, he kept improving the shipyard by every means to meet the continual expansion of his business. Over 500 vessels, both cruisers and gunboats for the Admiralty, as well as cargo and passenger steamers for private firms, were turned out under his superintendence. He also took an active part in municipal and political affairs, and, although unsuccessful in his contest for the borough against Mr. Isaac Wilson in 1885, he nevertheless contributed largely by his personal influence to the victory of his successor in the Conservative interest at the general election of 1900.

The honour of knighthood was conferred upon Sir Raylton in 1890, shortly after the visit of the Prince and Princess of Wales to Middlesbrough on the occasion of the opening of the new Town Hall and Municipal Buildings.

As a member of the Institution of Naval Architects he will always be remembered for the activity and geniality with which he took part in its proceedings, as well as for his great professional enthusiasm and ability. Joining in 1868, Sir Raylton was elected on the Council of the Institution in 1888. He was also a member of the Worshipful Company of Shipwrights; and served as a member of the Load Line Committee.

He was Deputy Lieutenant for the North Riding of Yorkshire, a Major of Artillery Volunteers, and a Knight of the Order of St. John of Jerusalem.

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## CARL FERDINAND LAEISZ.

THE untimely death, at the early age of forty-seven, of Herr Carl Ferdinand Laeisz deprives the Institution of one who did much to promote cordial relations with the leaders of shipping and allied industries on the Continent.

Born in Hamburg in 1853, Herr Laeisz was first prepared for a mercantile career in his native town, but after completing his studies, he made an extended trip round the world, in 1875 to 1877, and, on his return, was taken into partnership by his grandfathers in the firm of F. Laeisz & Co. In 1881 he was made manager of the Hansa Shipping Company, which was eventually absorbed by the Hamburg-American Line.

The great work of his life, however, was the creation of the German Maritime Association, at the head of which he remained from its foundation in 1887 until his death. From 1884 Herr Laeisz was a member of the Hamburg Chamber of Commerce, and he filled the office of President from 1895 to 1898.

His connection with the Institution of Naval Architects dates from 1896, when he was elected an Associate, and he showed his interest in its proceedings by frequent attendance at the Summer Meetings, and by the part he took in discussions upon topics of interest to shipowners.

When the Institution held its Summer Meeting in Hamburg and Berlin in 1896, it was Herr Laeisz who, in company with Herr Otto Schlick, welcomed the members on board the *Fürst Bismarck*, and the success of those meetings was largely due to Herr Laeisz's untiring exertions and genial hospitality.

His death took place on August 22, 1900, after an illness of several months.

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# IOWA.



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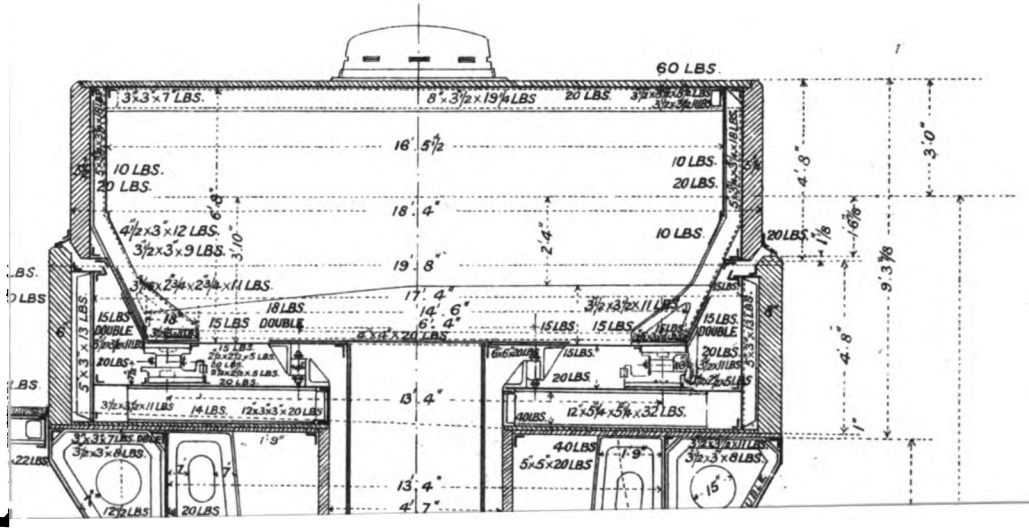


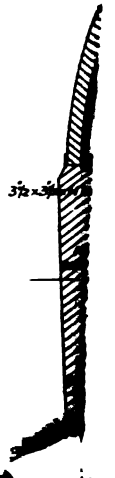

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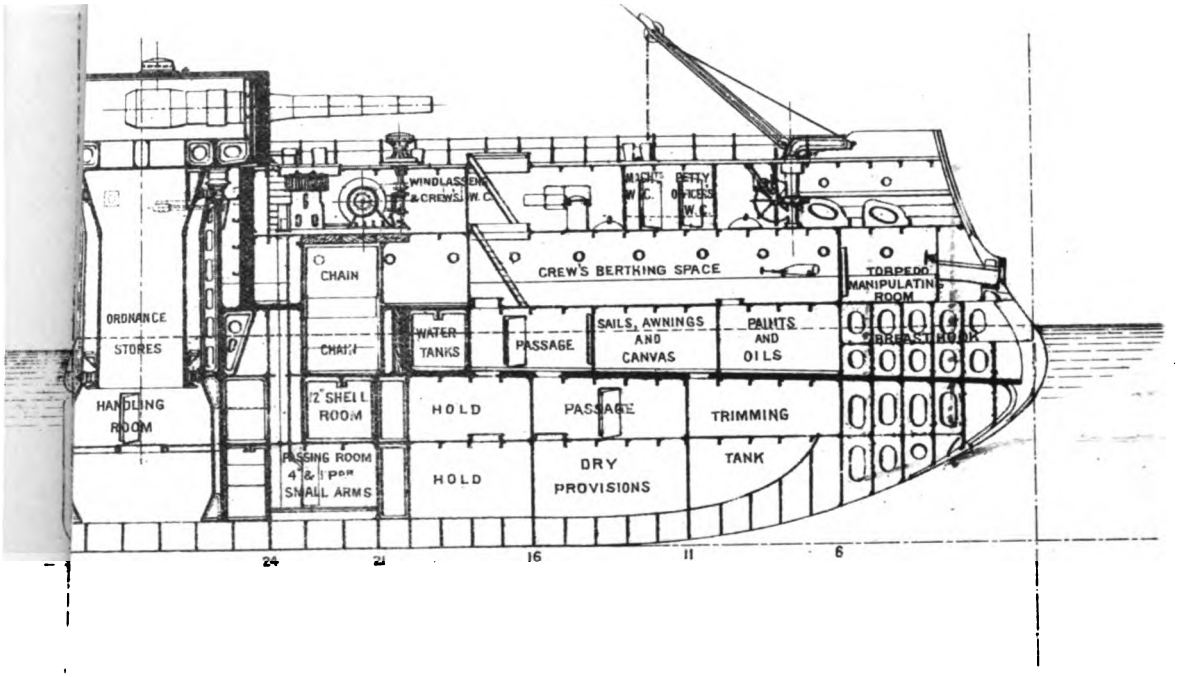


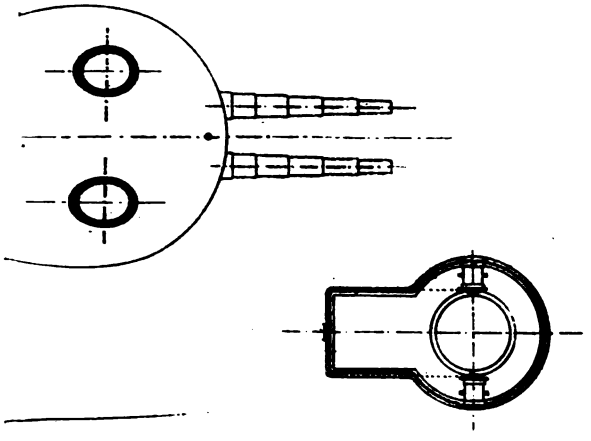

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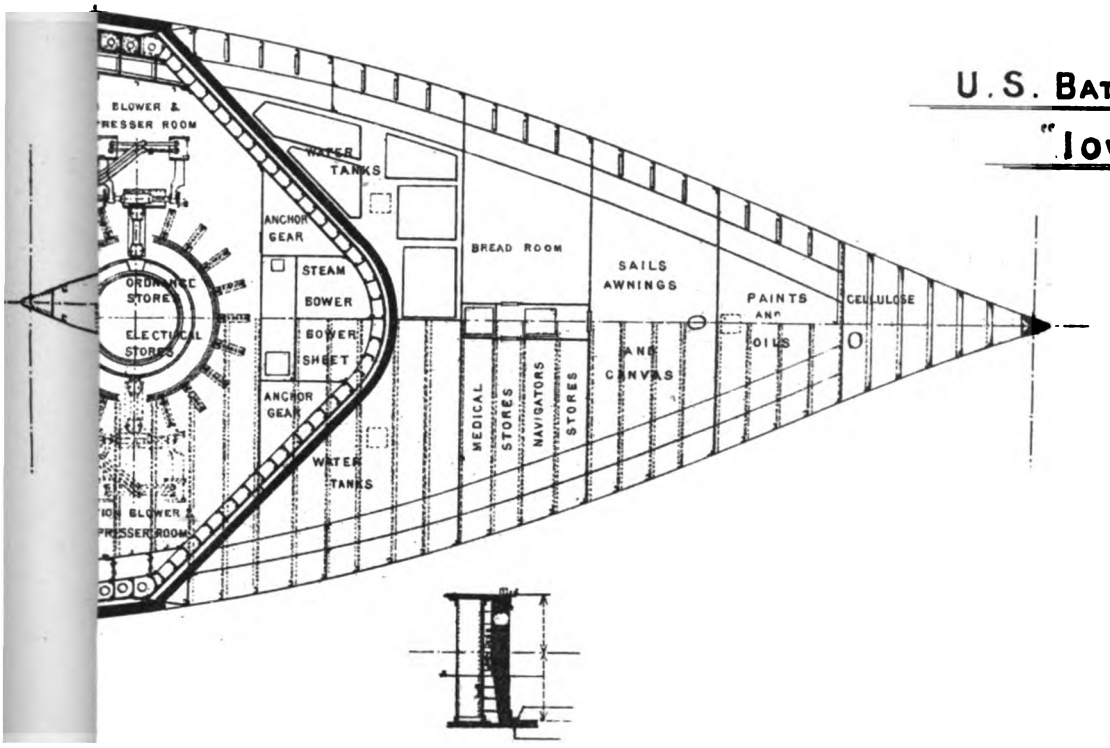


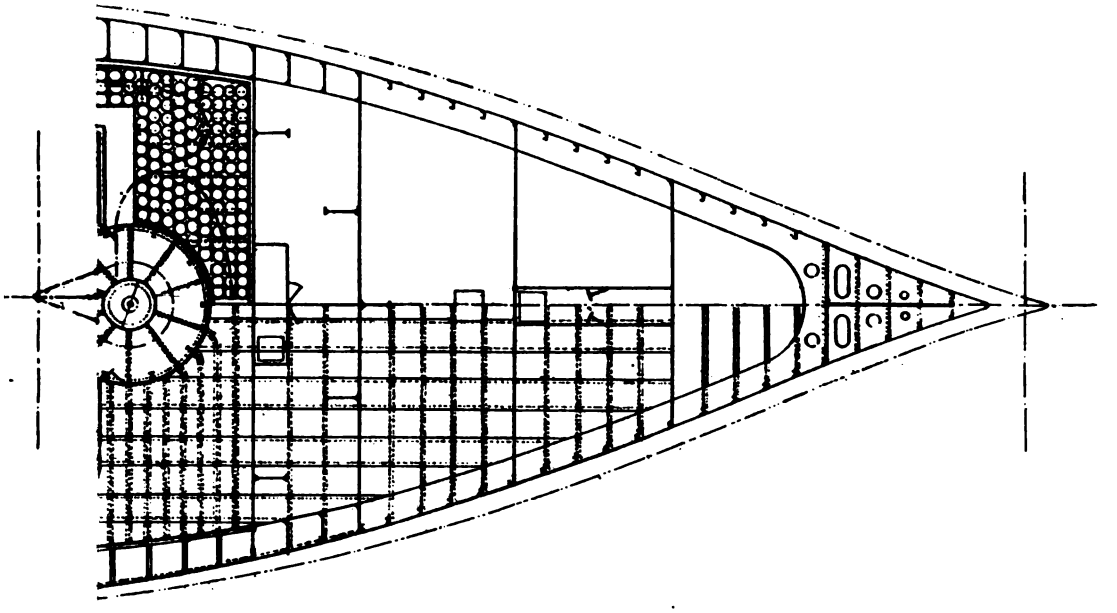

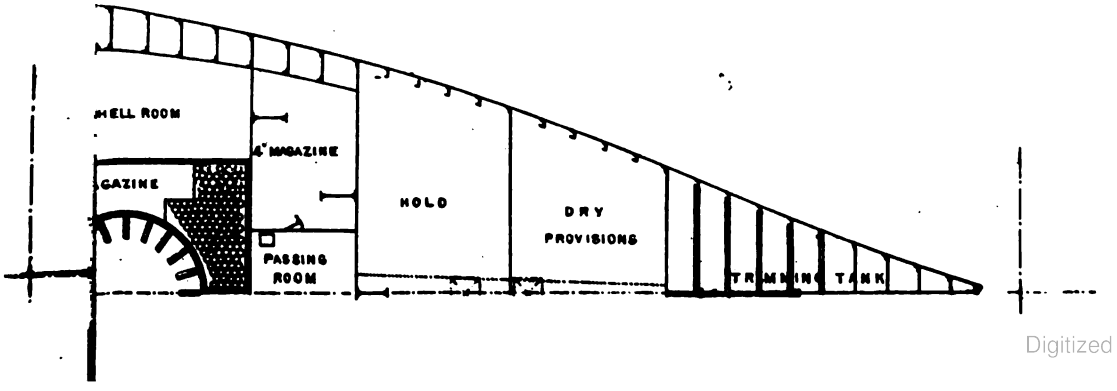


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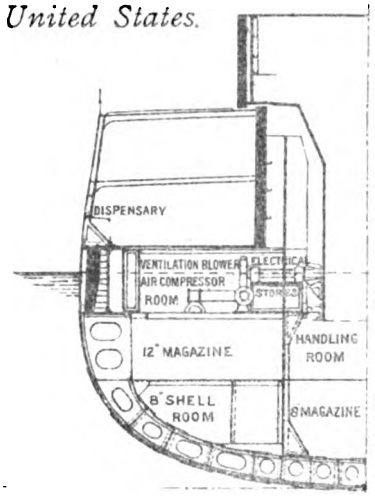


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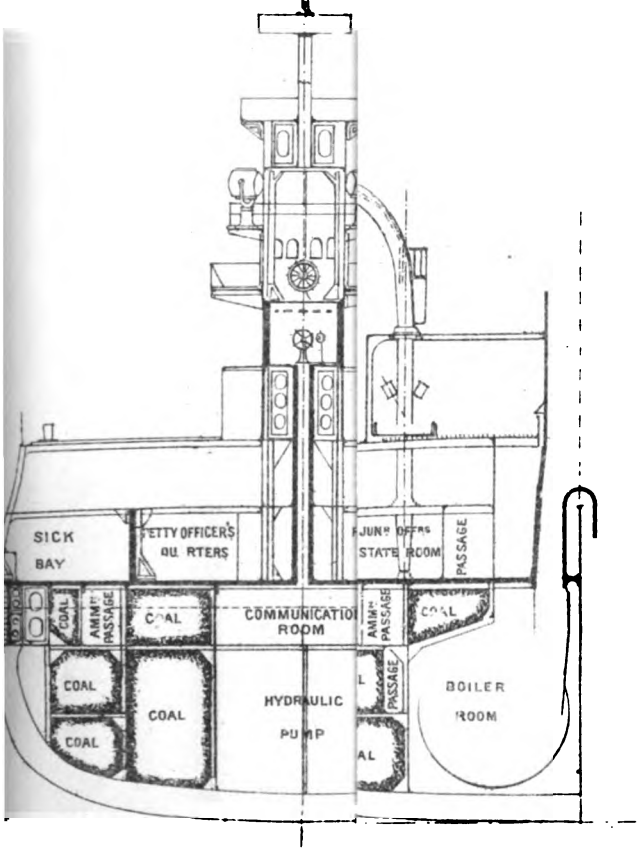


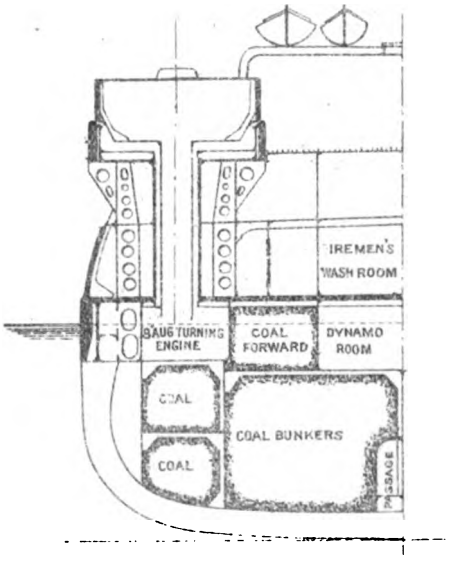


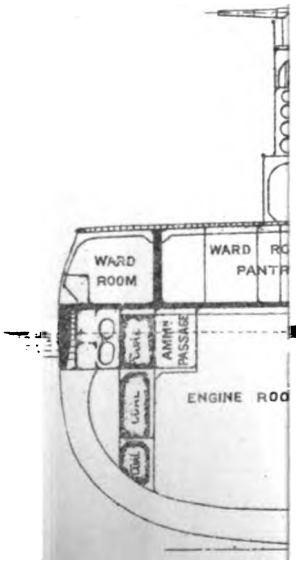


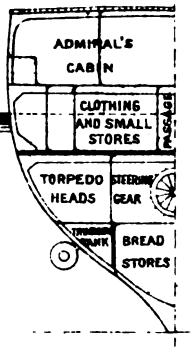


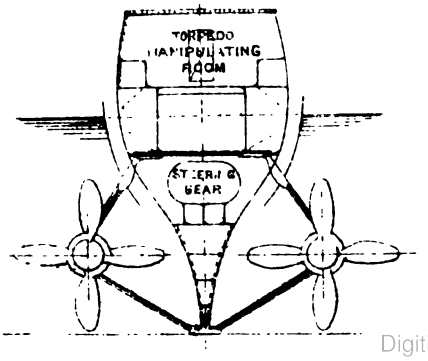


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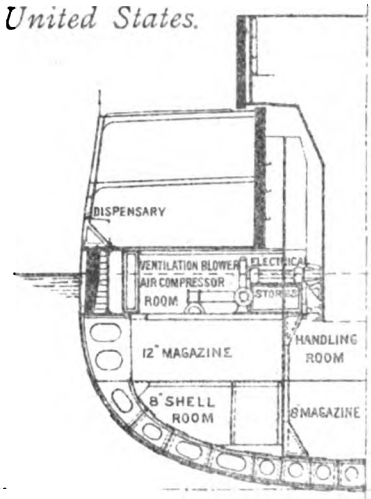


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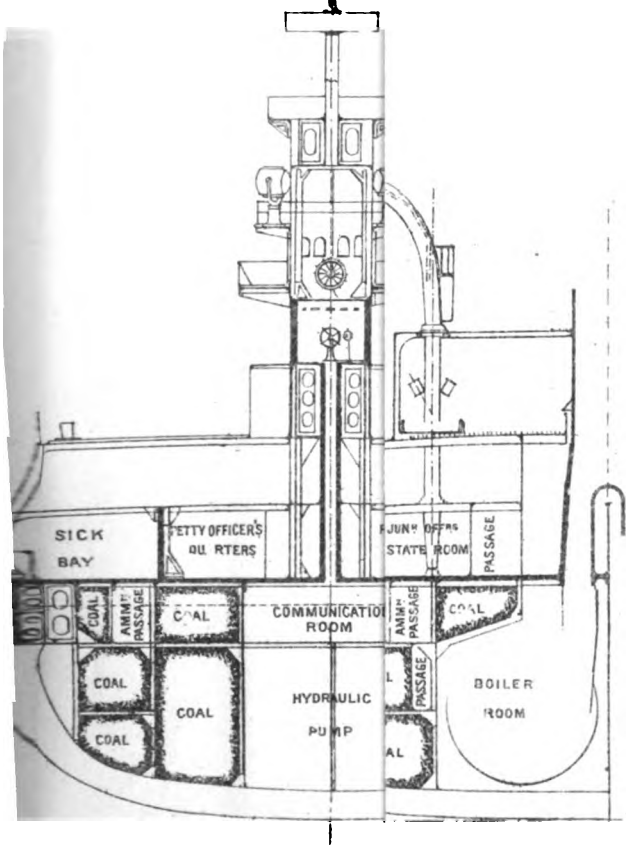


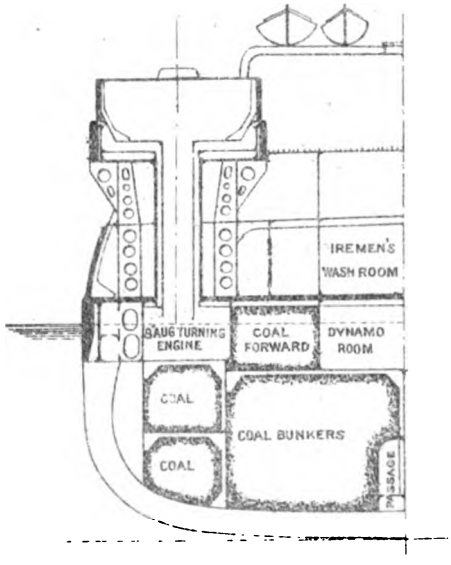


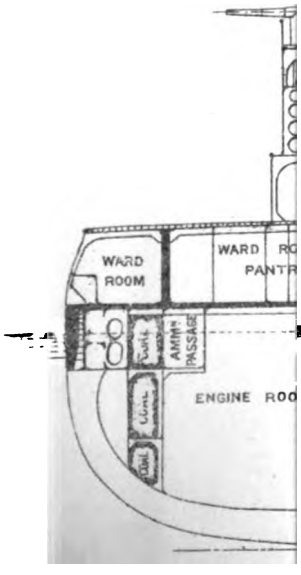


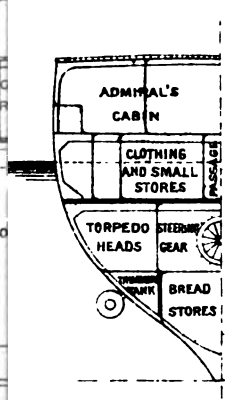


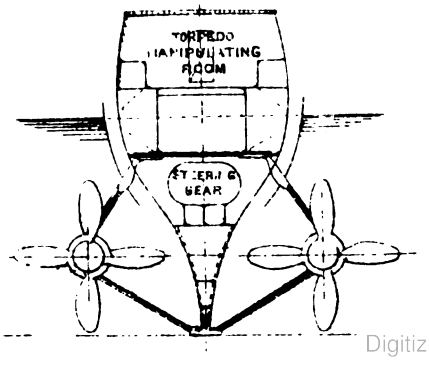


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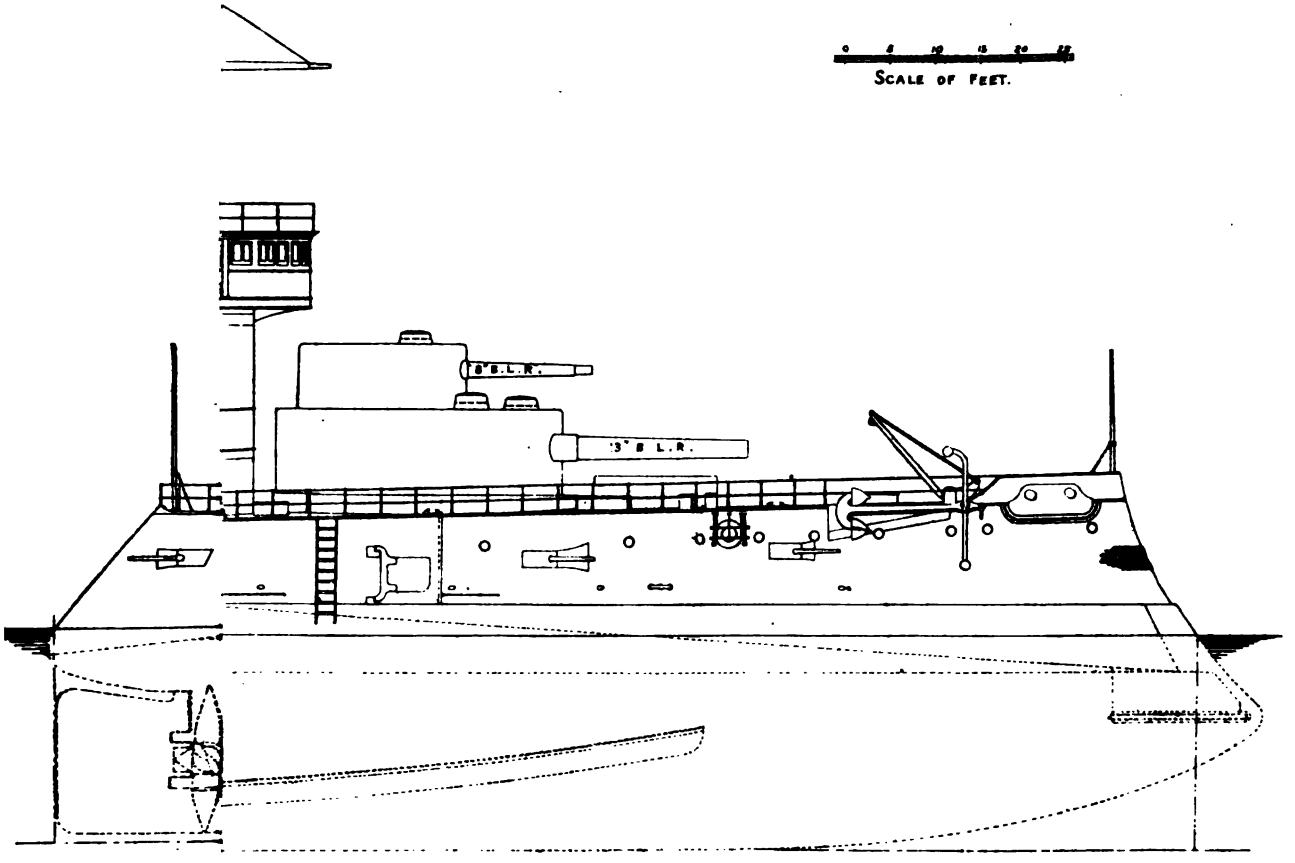


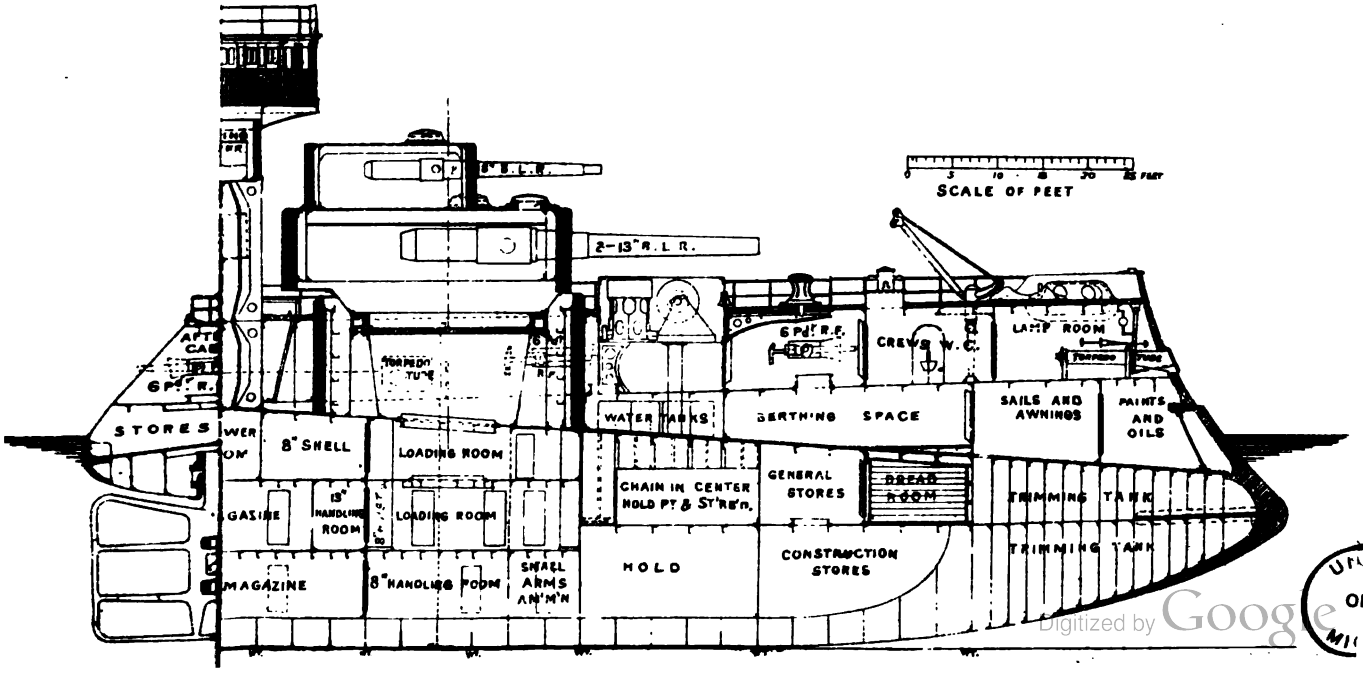


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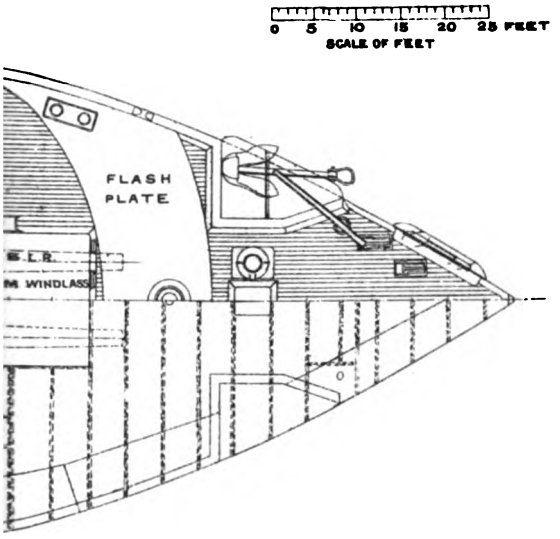


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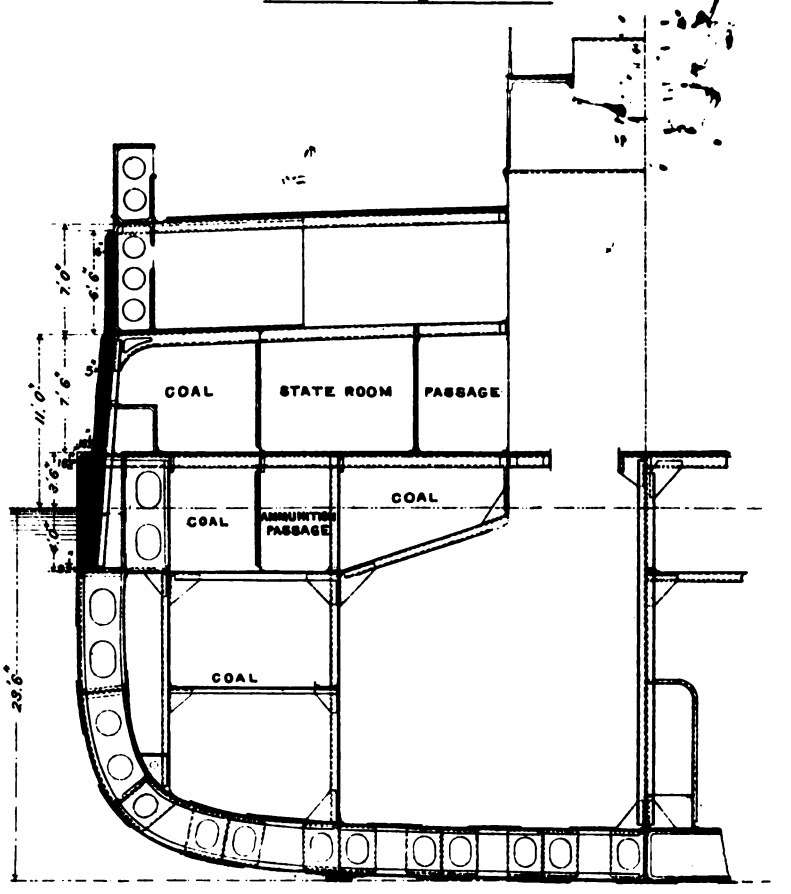


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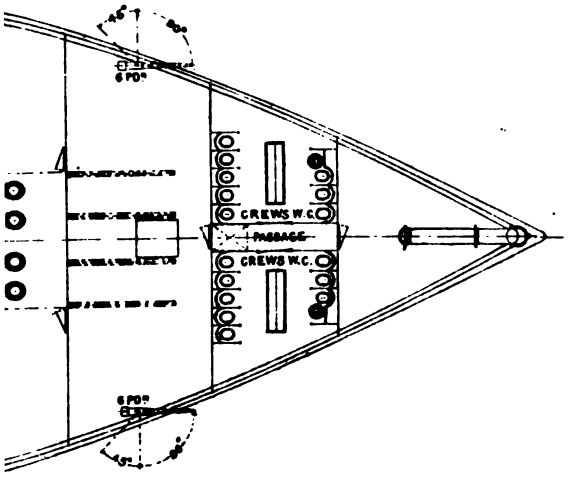


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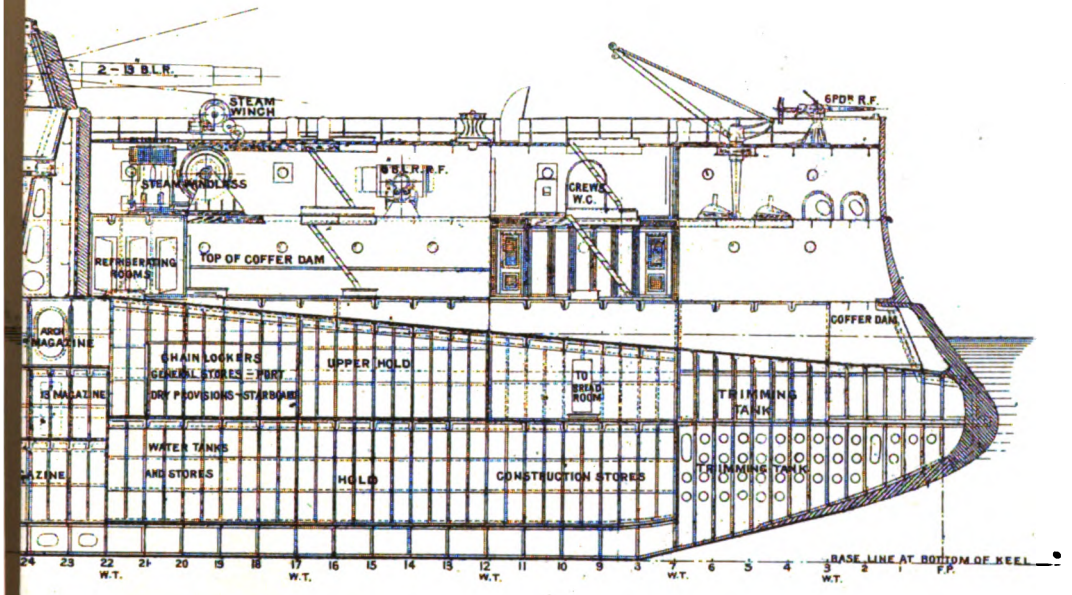


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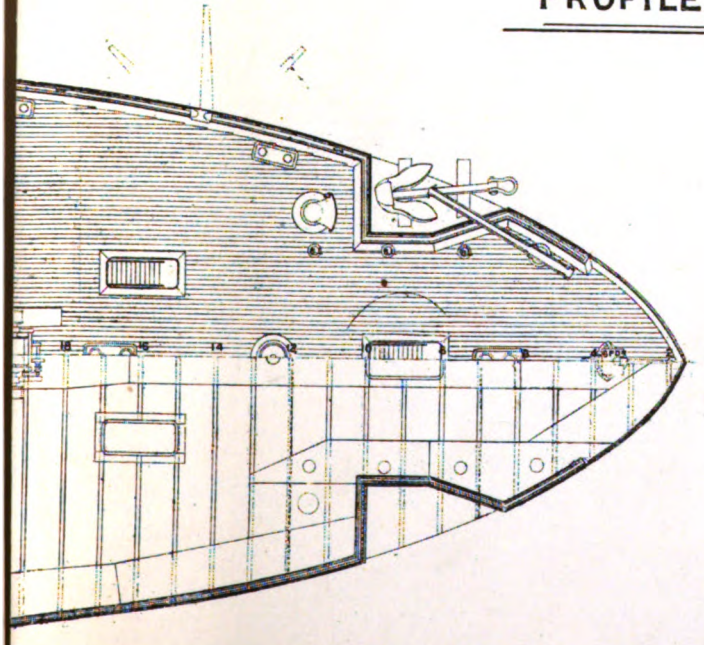

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# U. S. BATTLESHIPS.

## ALABAMA AND CLASS

## ALABAMA, ILLINOIS, WINCONSIN.

### BERTH DECK

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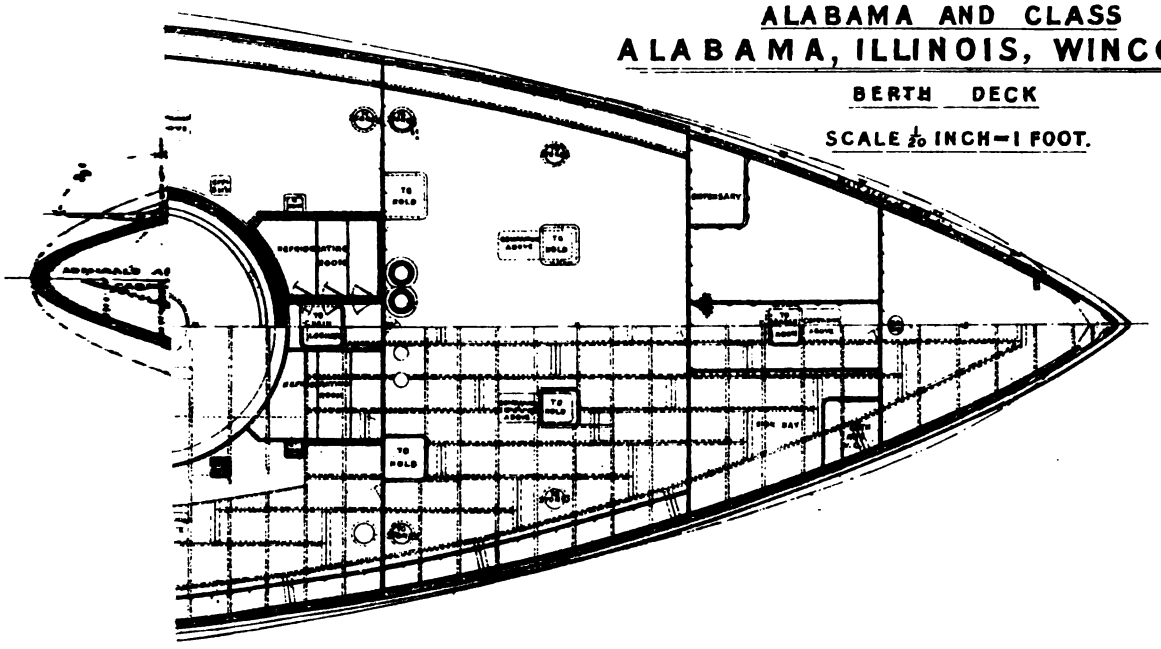


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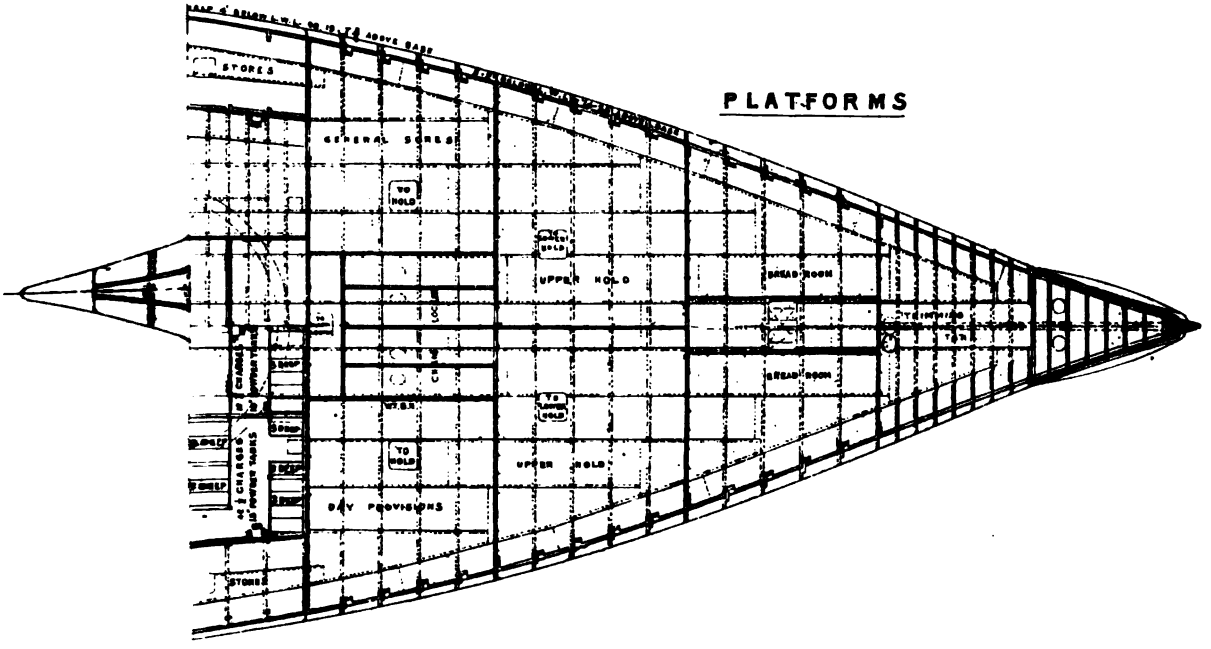


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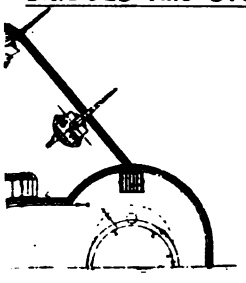

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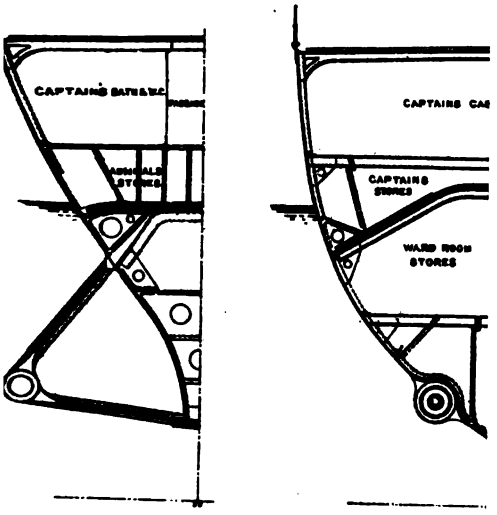
*To Illustrate Q*

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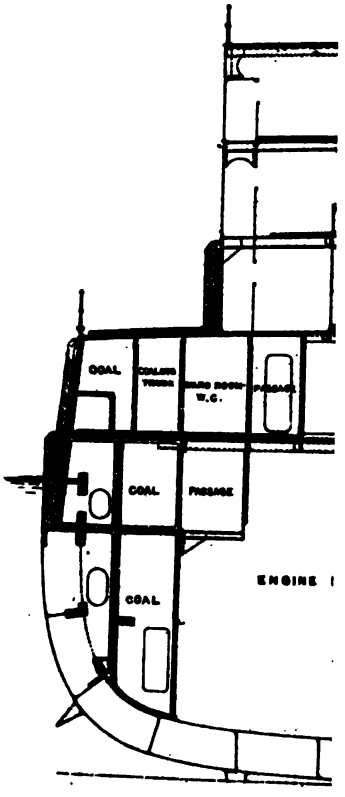


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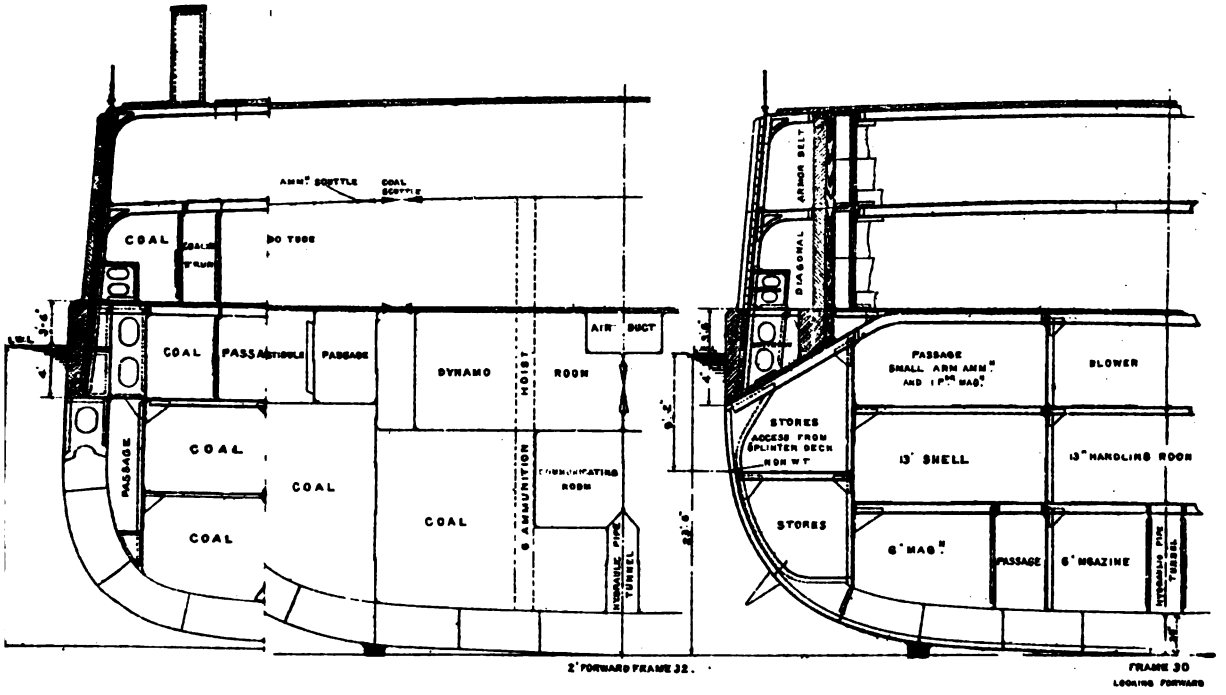
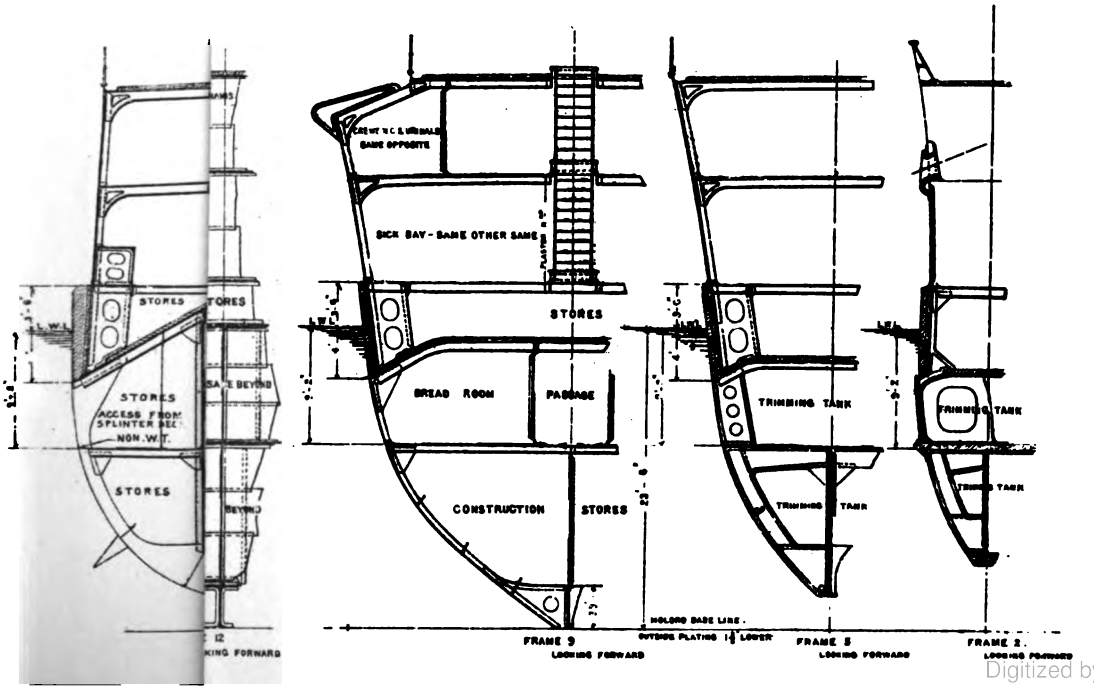
# BATTLESHIPS "ALABAMA" ILLINOIS "WISCONSIN".

## FORWARD SECTIONS

SCALE OF DRAWING $\frac{1}{16}$ INCH 1 FOOT.

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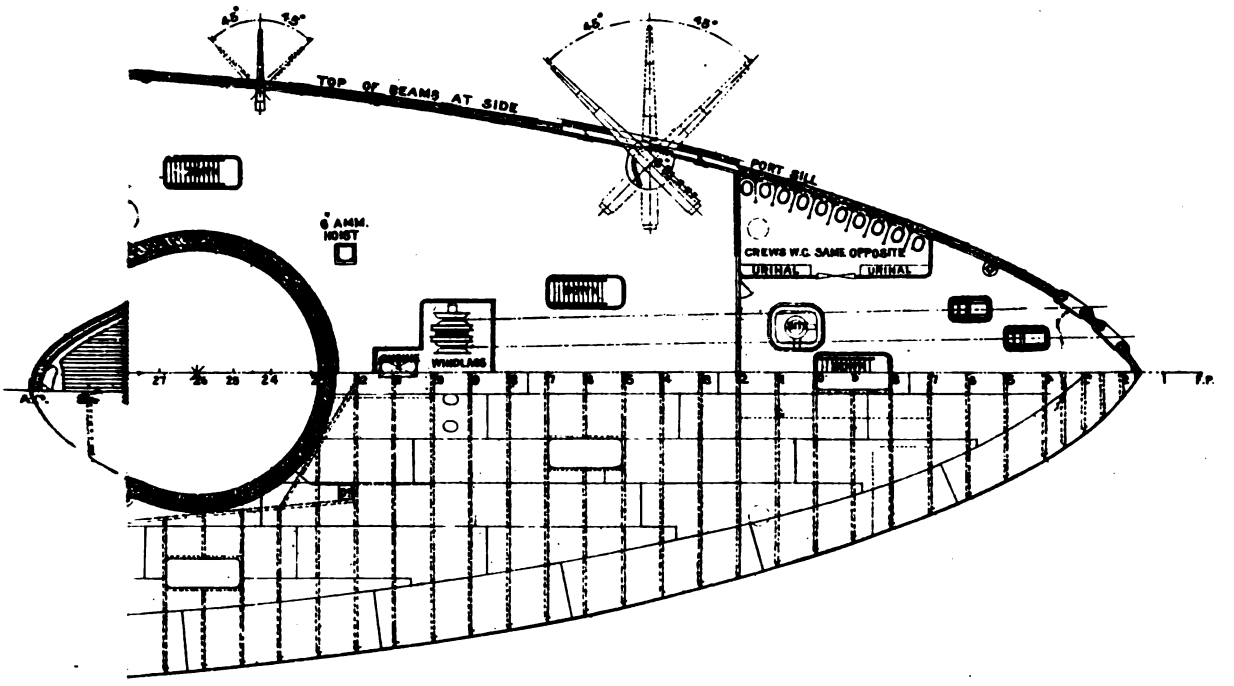
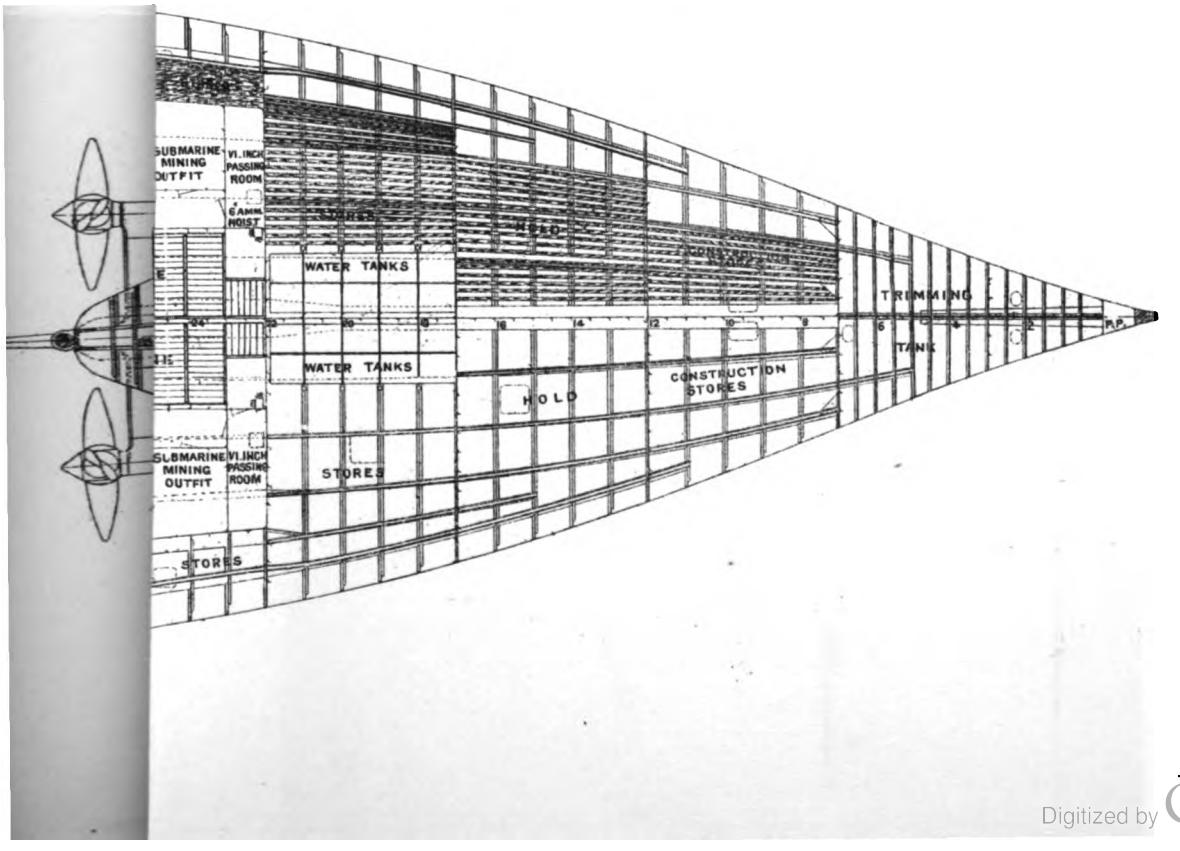
# U. S. BATTLESHIPS, ALABAMA. ILLINOIS. WISCONSIN.

## MAIN DECK.




## HOLD AND MAGAZINES.

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*val Construction in the United States.*

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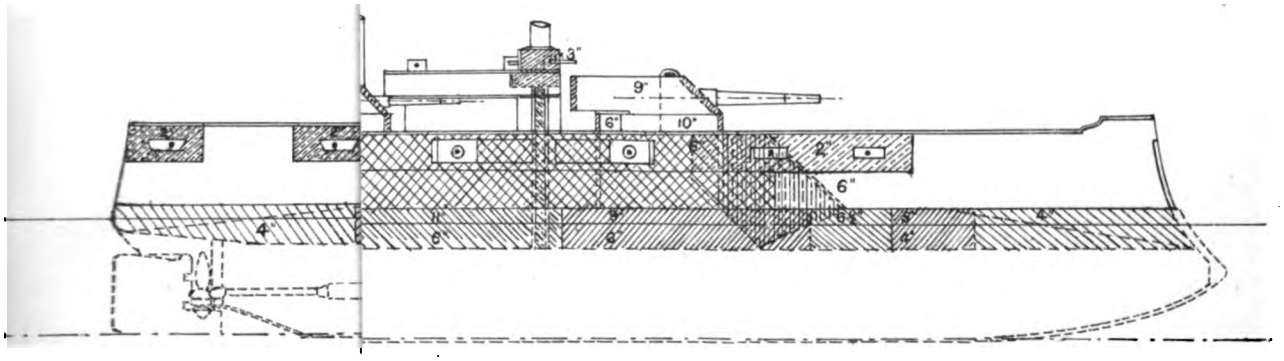
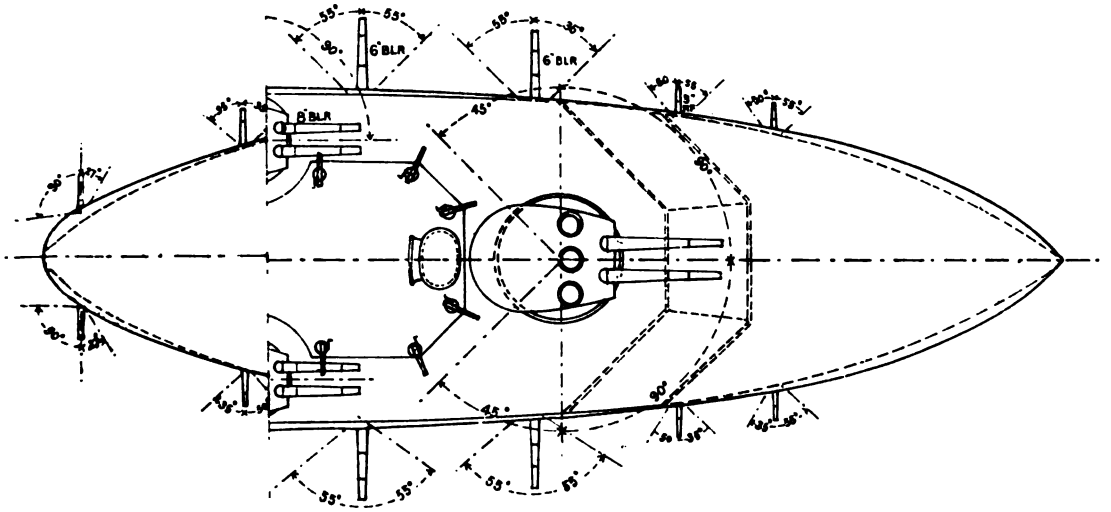
K GUNS.

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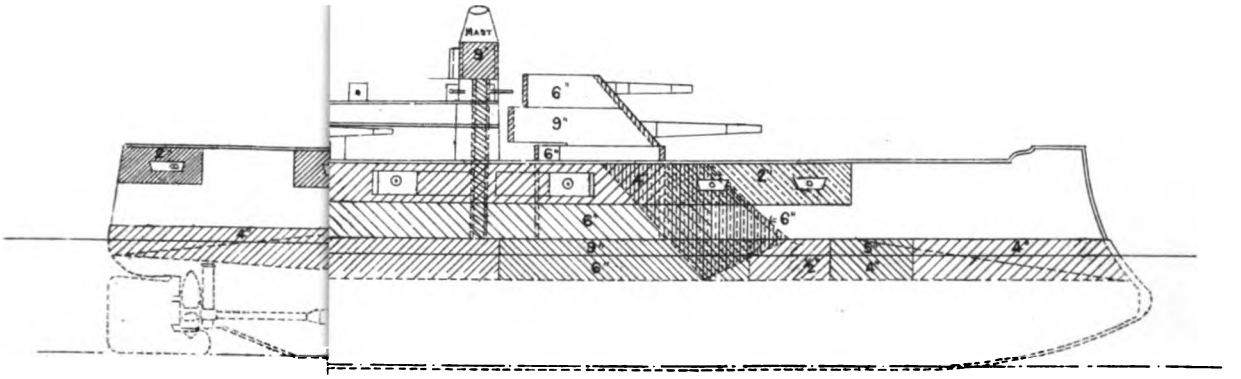
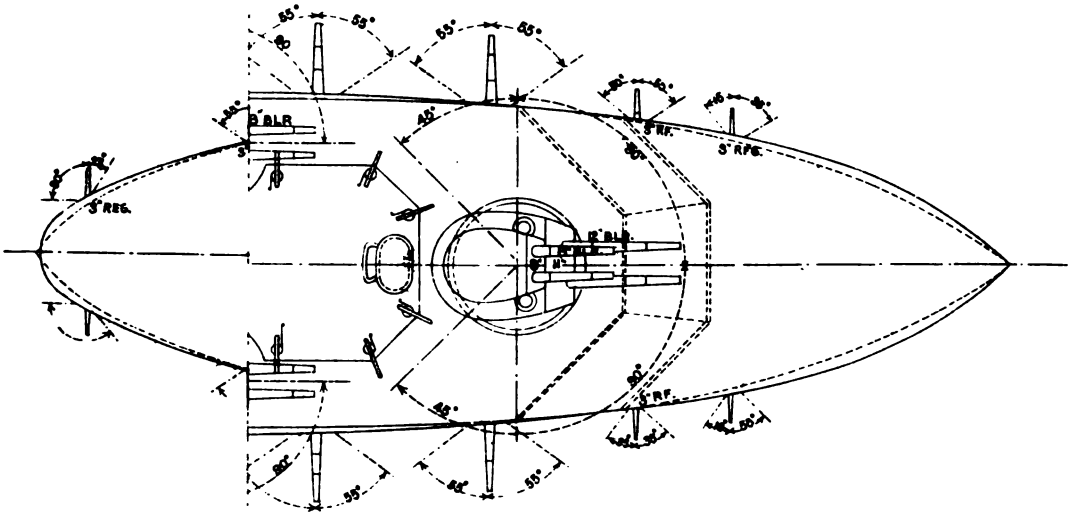
onstruction in the United States.

A.

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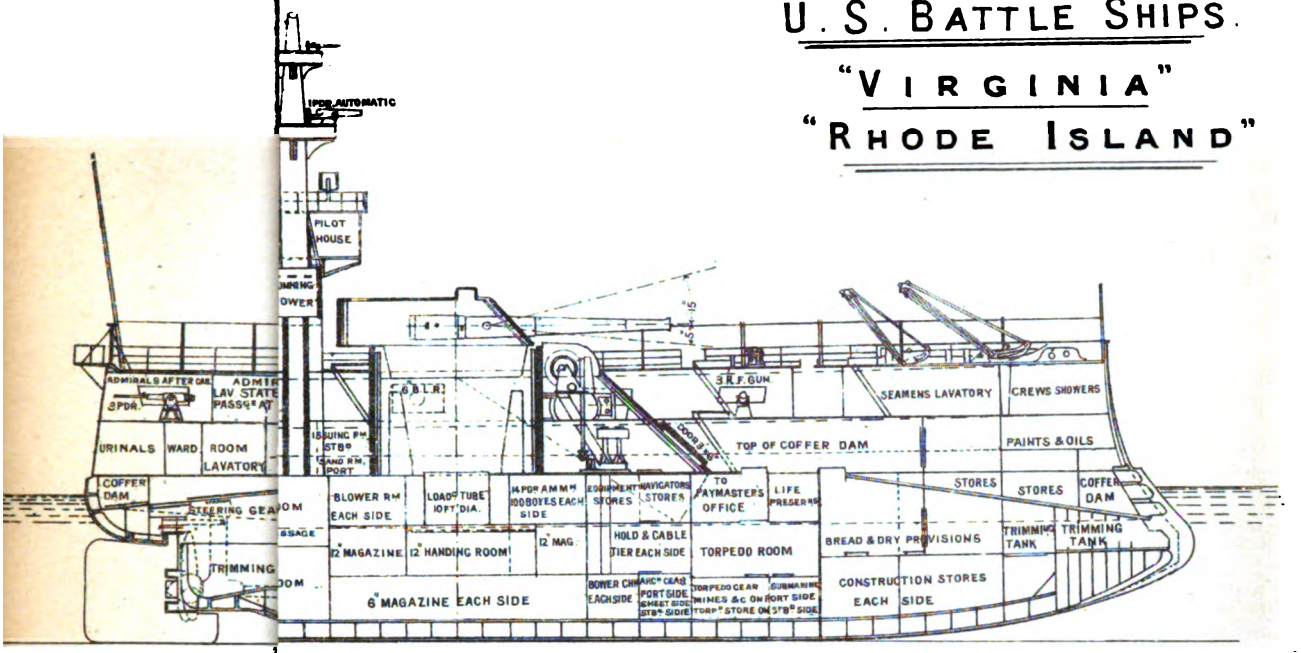
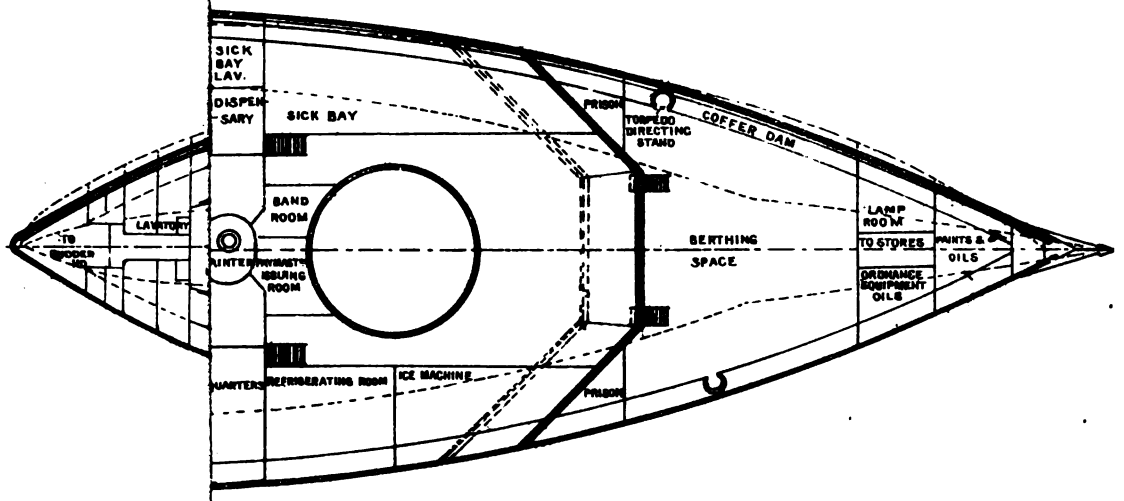
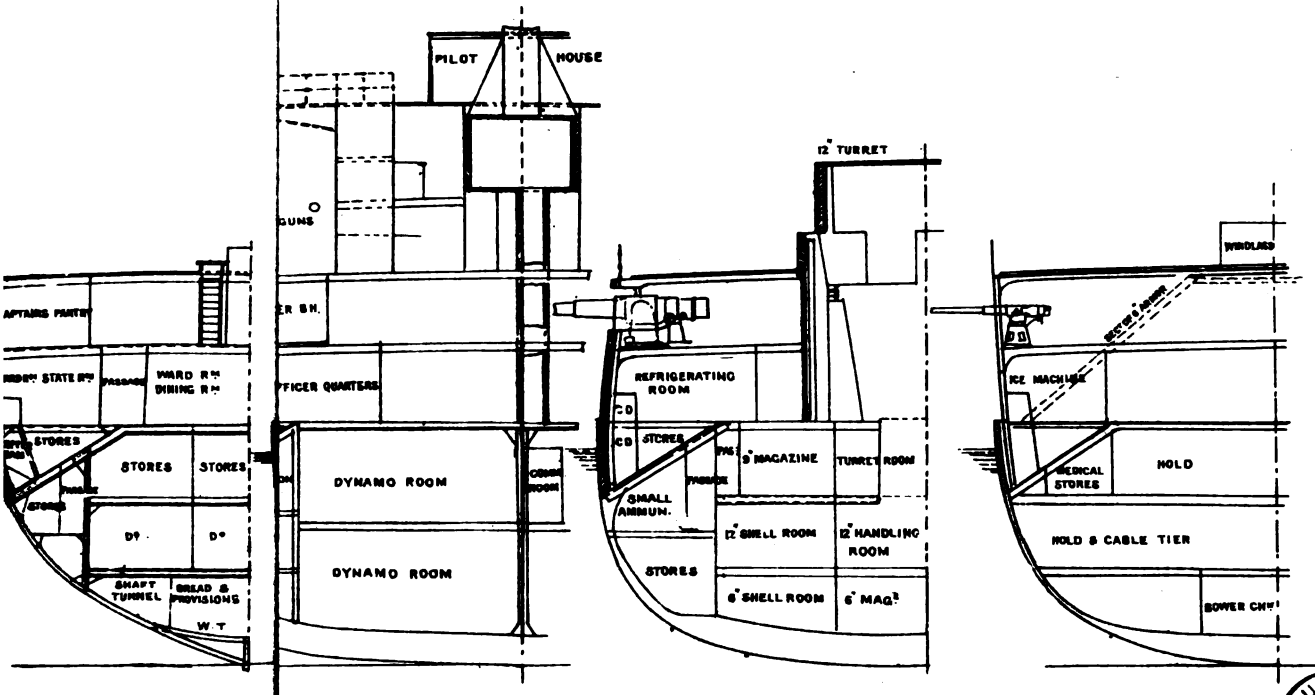
# U. S. BATTLE SHIPS.

## "VIRGINIA"
## "RHODE ISLAND"

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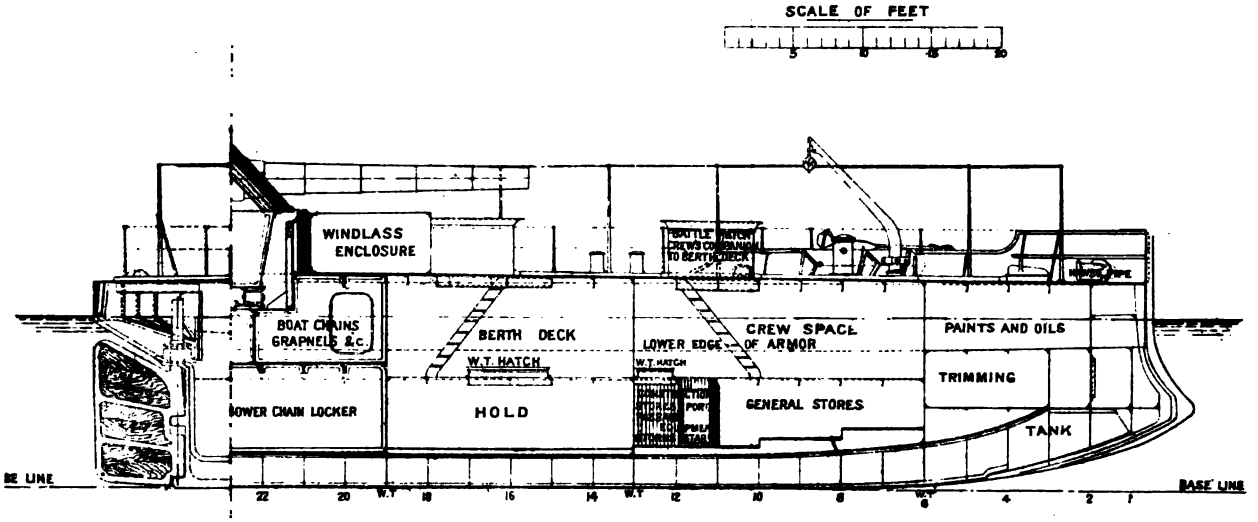
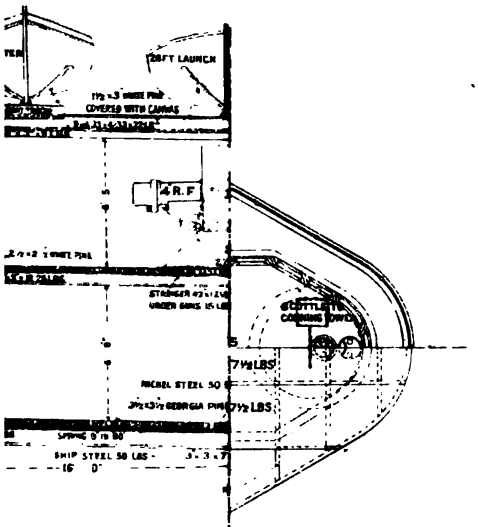
# ARKANSAS AND CLASS.

## PROFILE INBOARD.

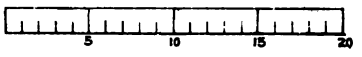



## IDSHIP SECTIO




## BRIDGE DECK.

SCALE OF FEET.

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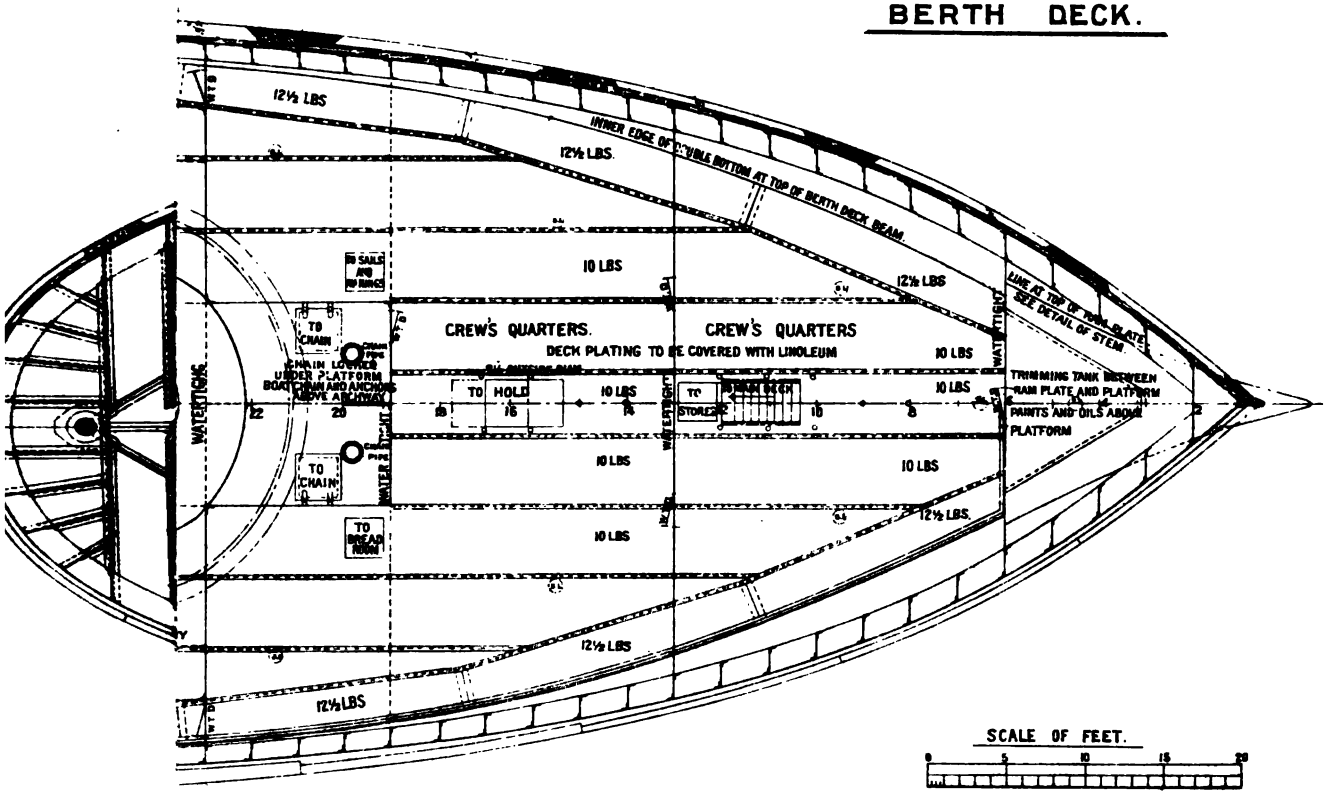
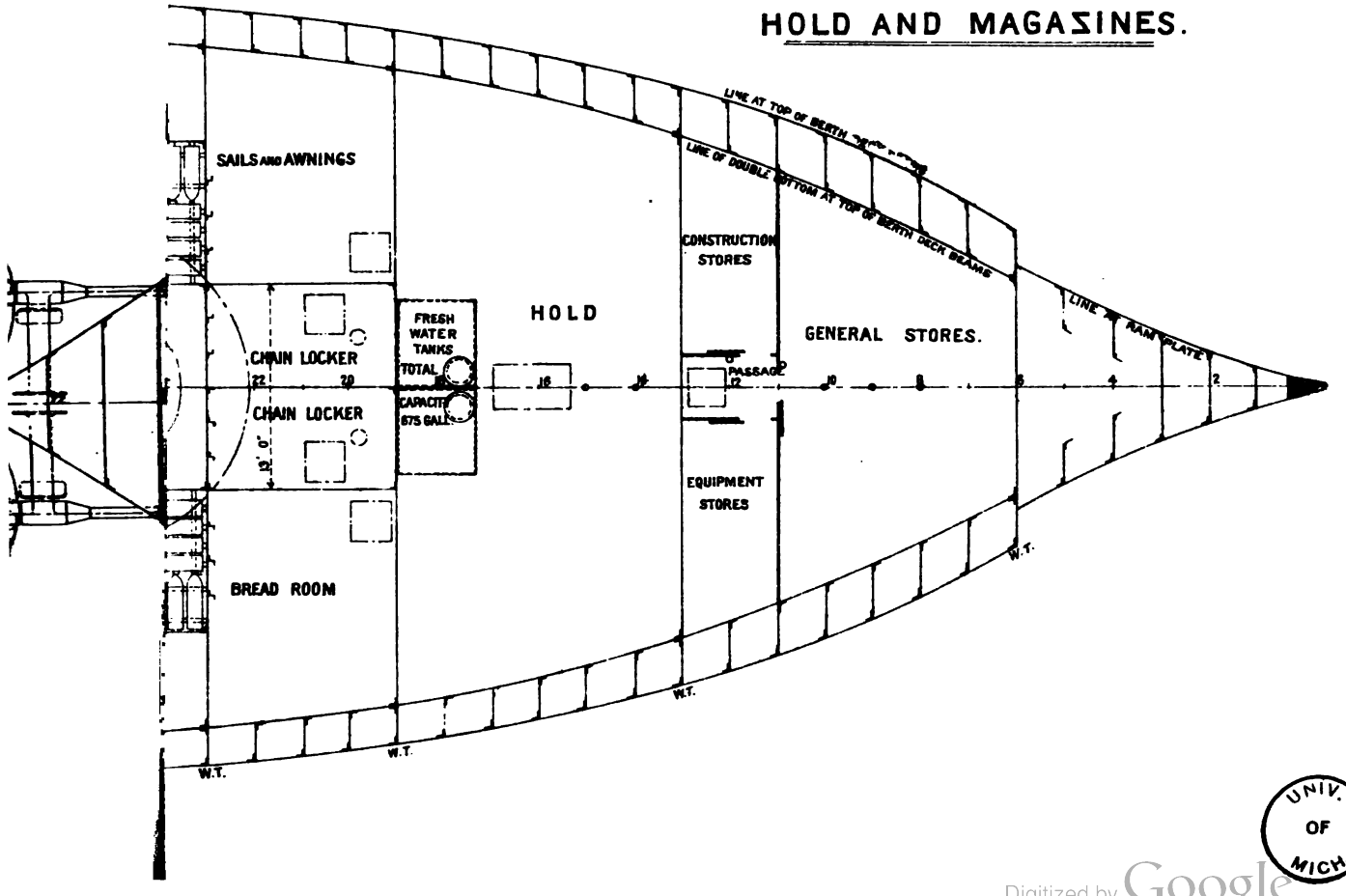
# ARKANSAS AND CLASS.

## BERTH DECK.




## HOLD AND MAGASINES.

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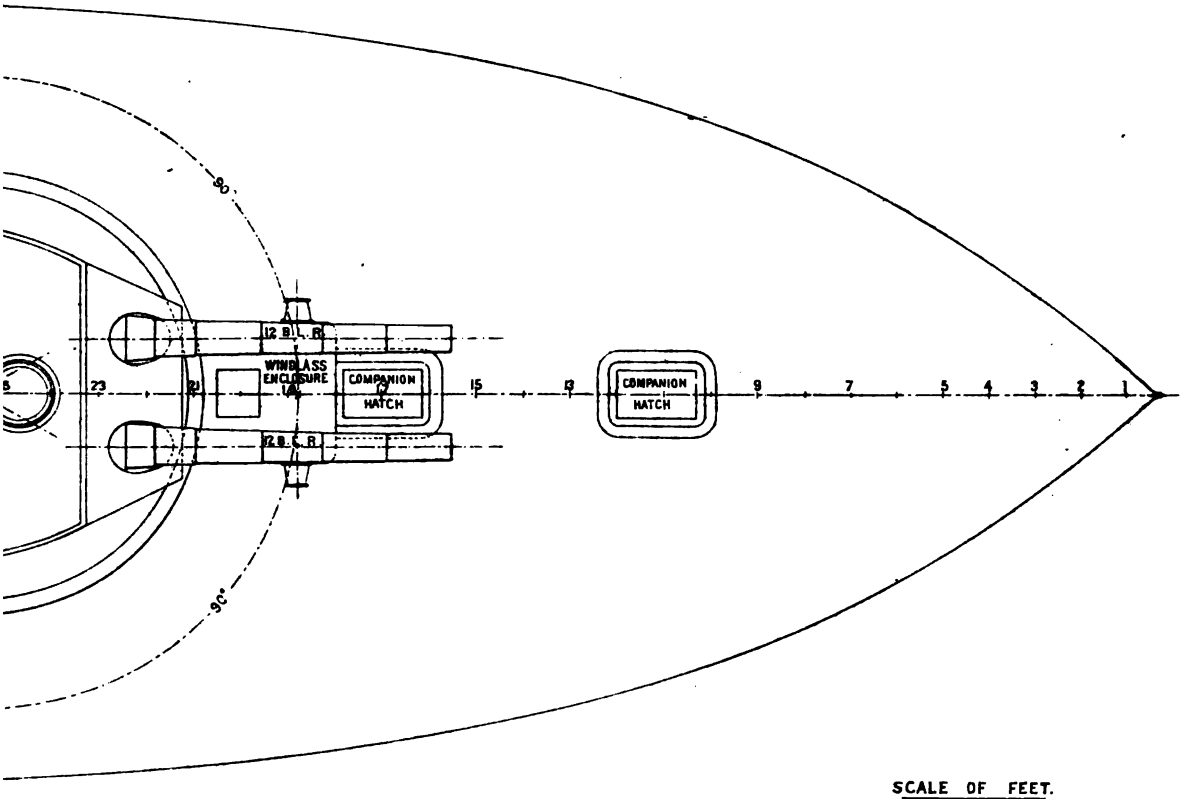
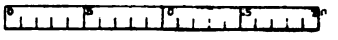
# ARKANSAS AND CLASS.

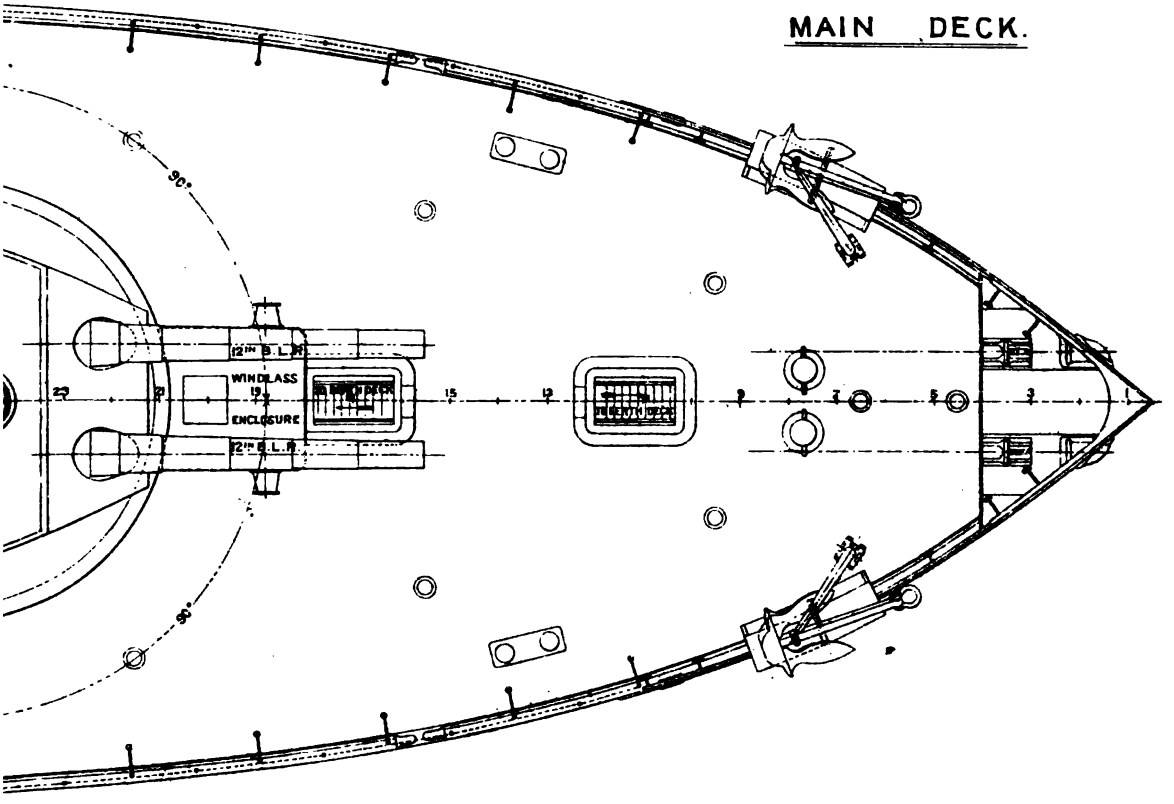
## SUPERSTRUCTURE DECK.




SCALE OF FEET.



## MAIN DECK.

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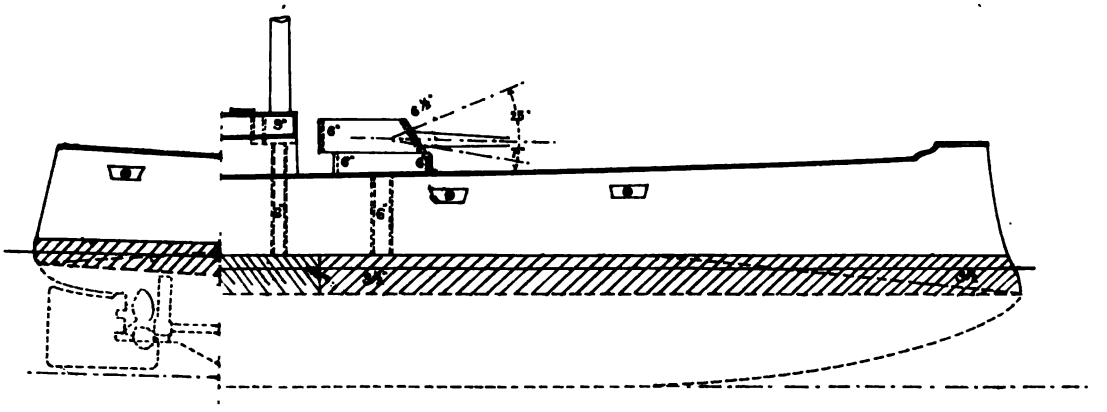
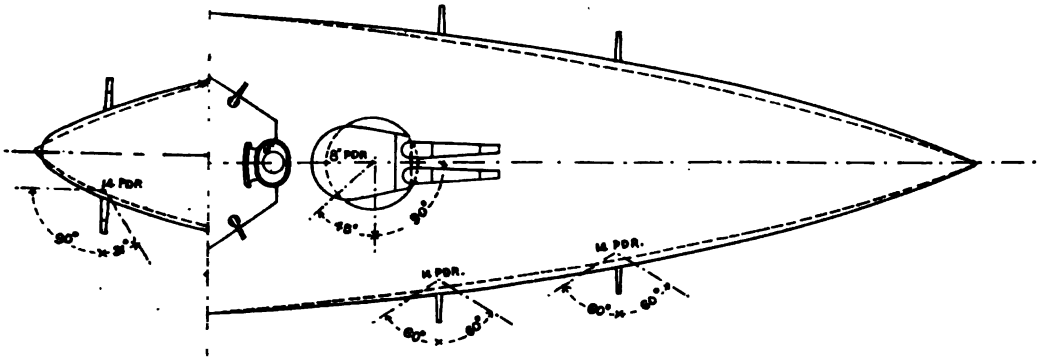
he United States.

AKOTA.



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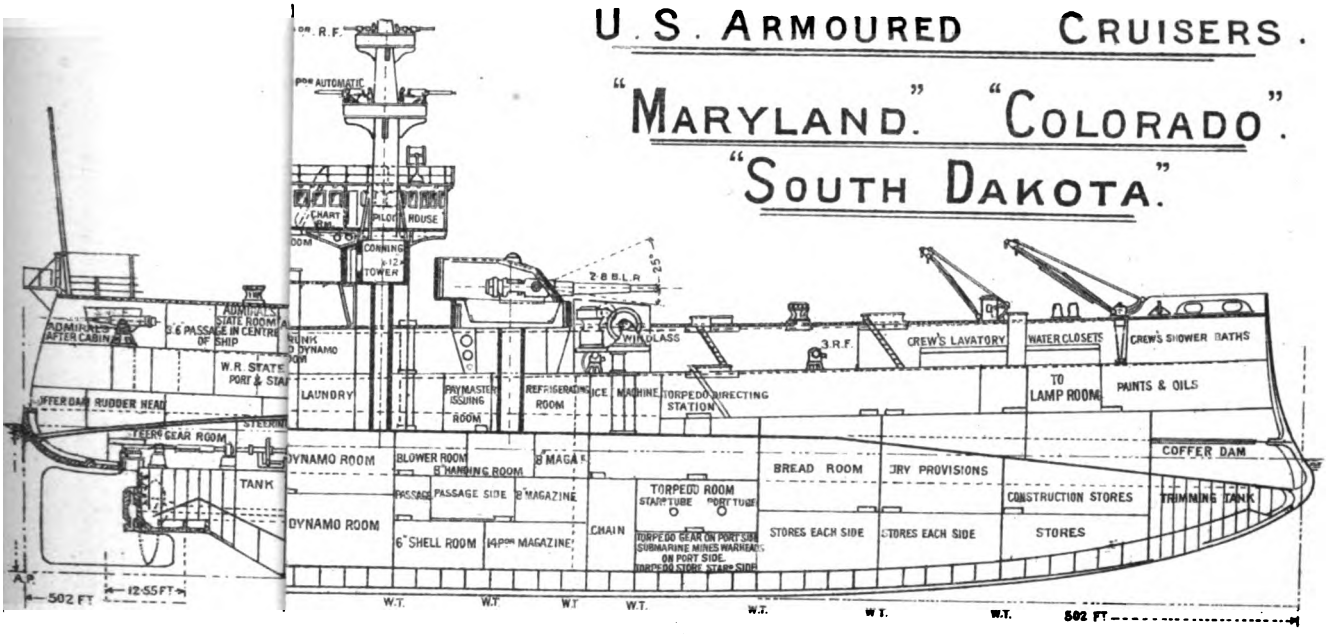
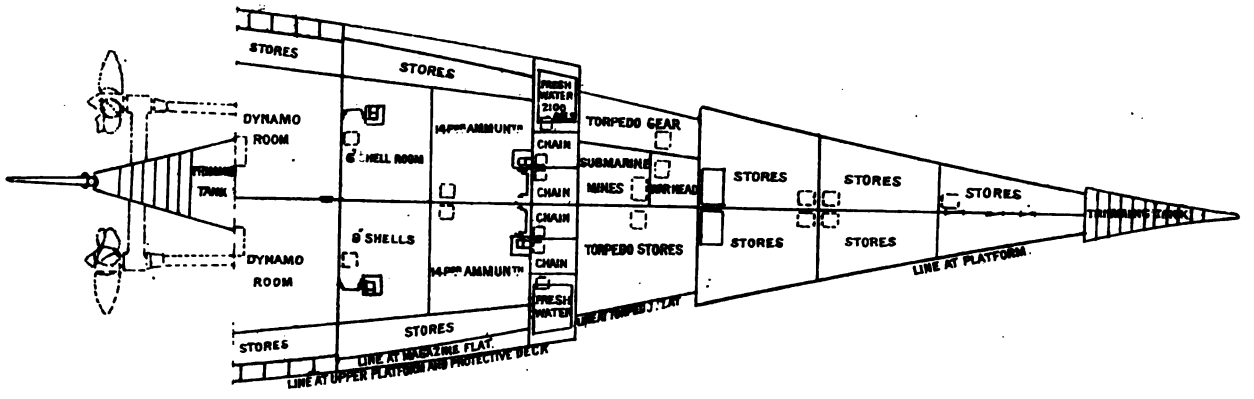
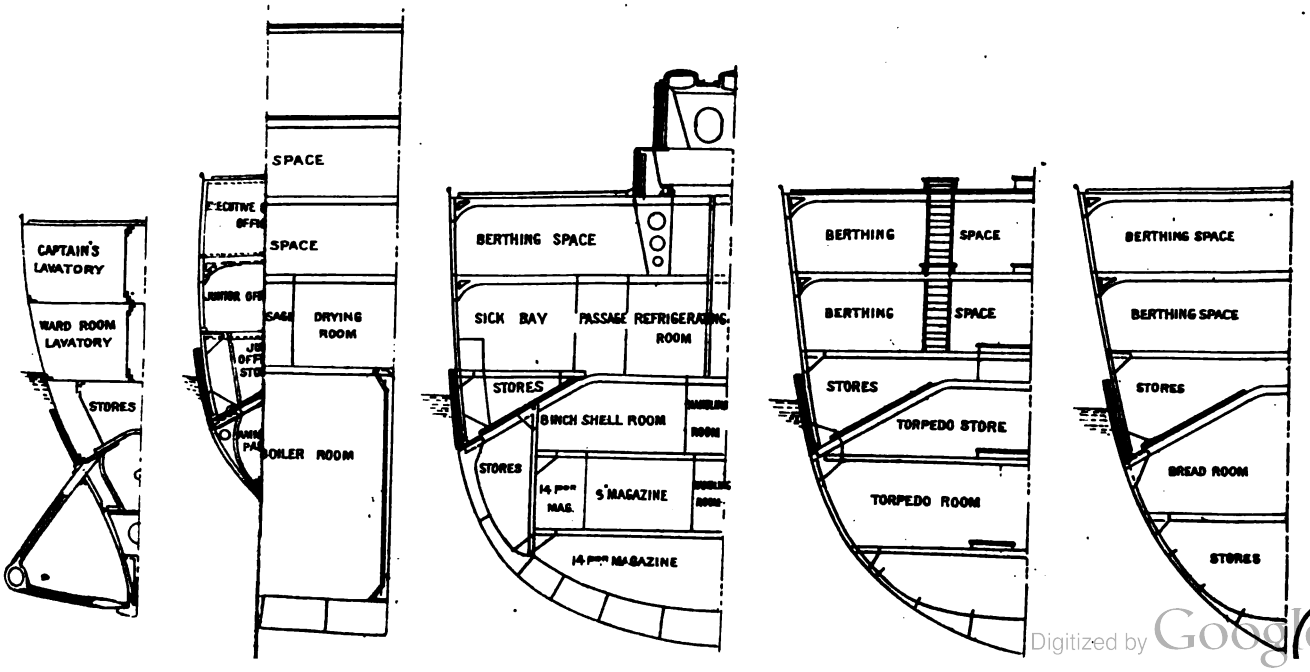
# U.S. ARMOURED CRUISERS.

## "MARYLAND." "COLORADO". "SOUTH DAKOTA."

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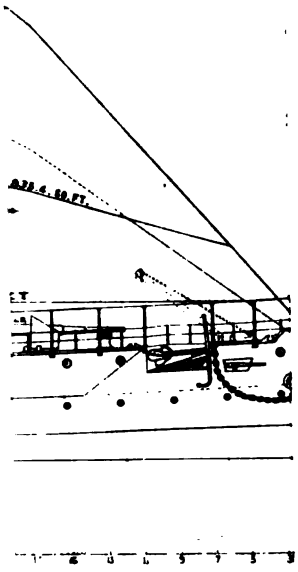
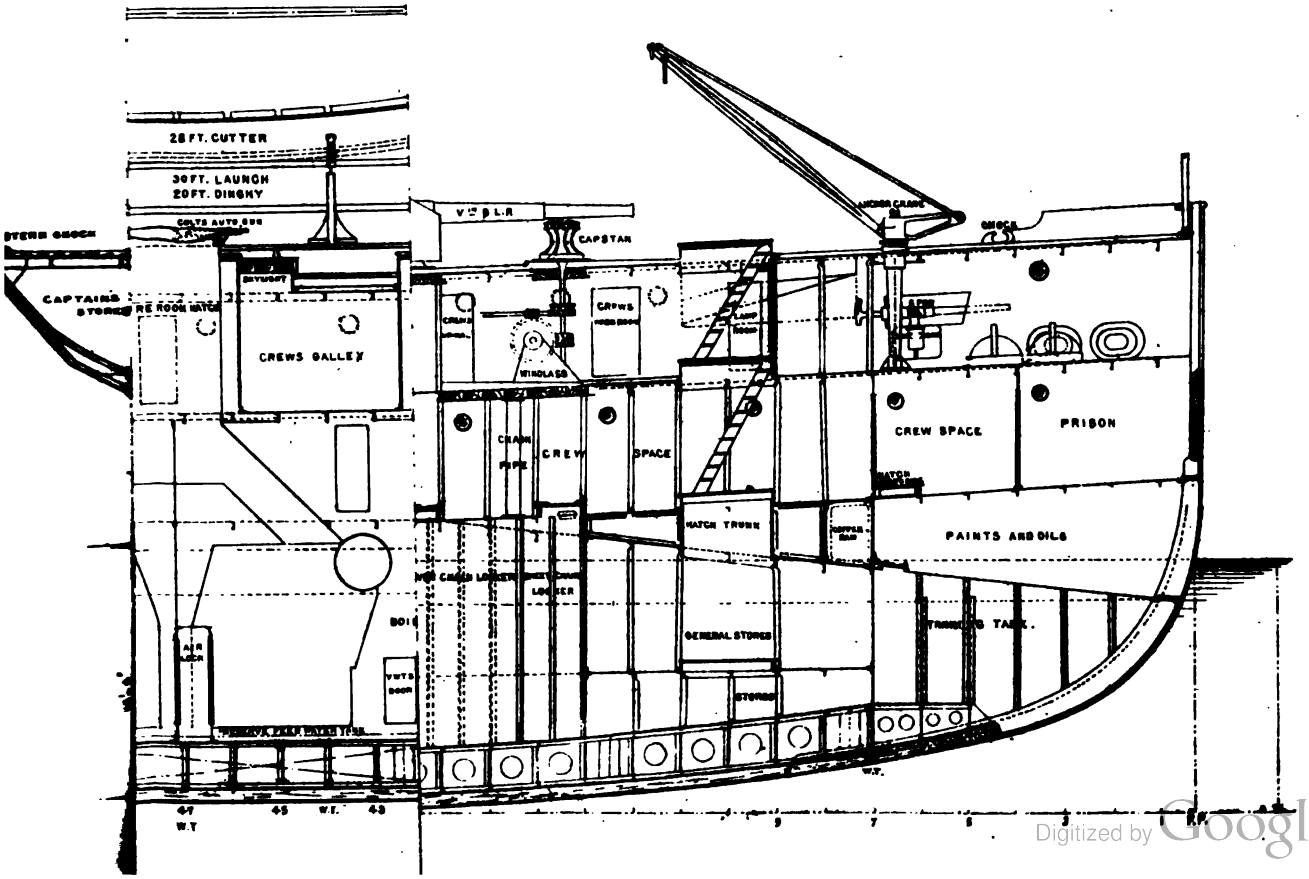
# S. PROTECTED CRUISERS.

## DENVER CLASS.

### SHEATHED CRUISERS.



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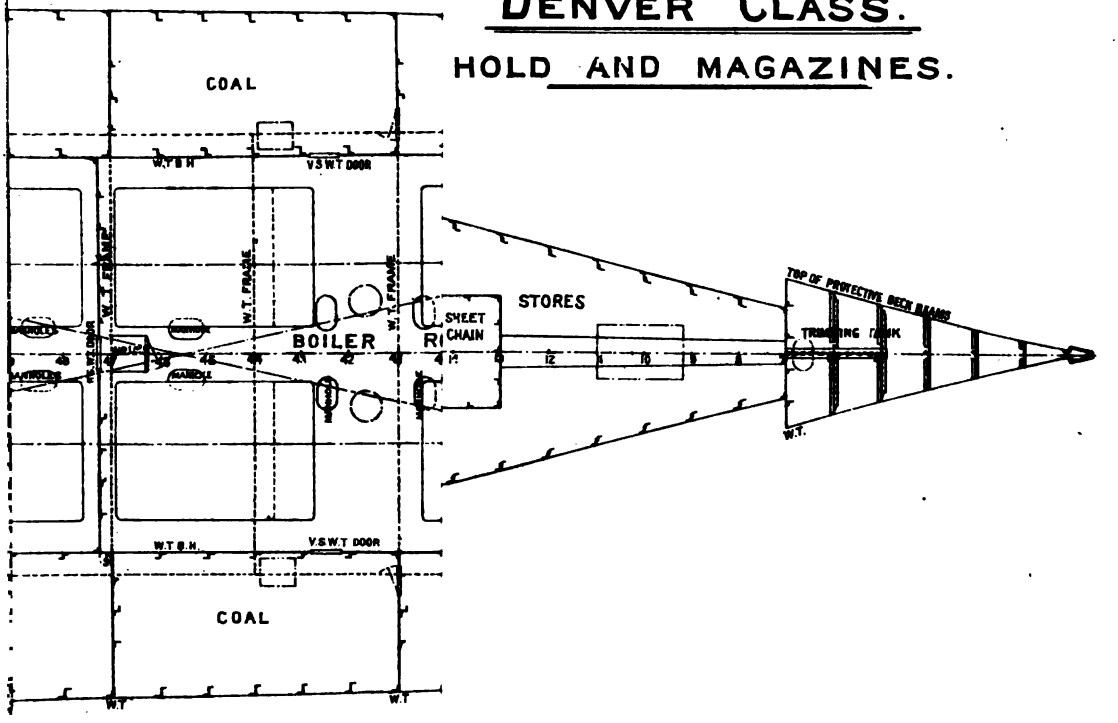
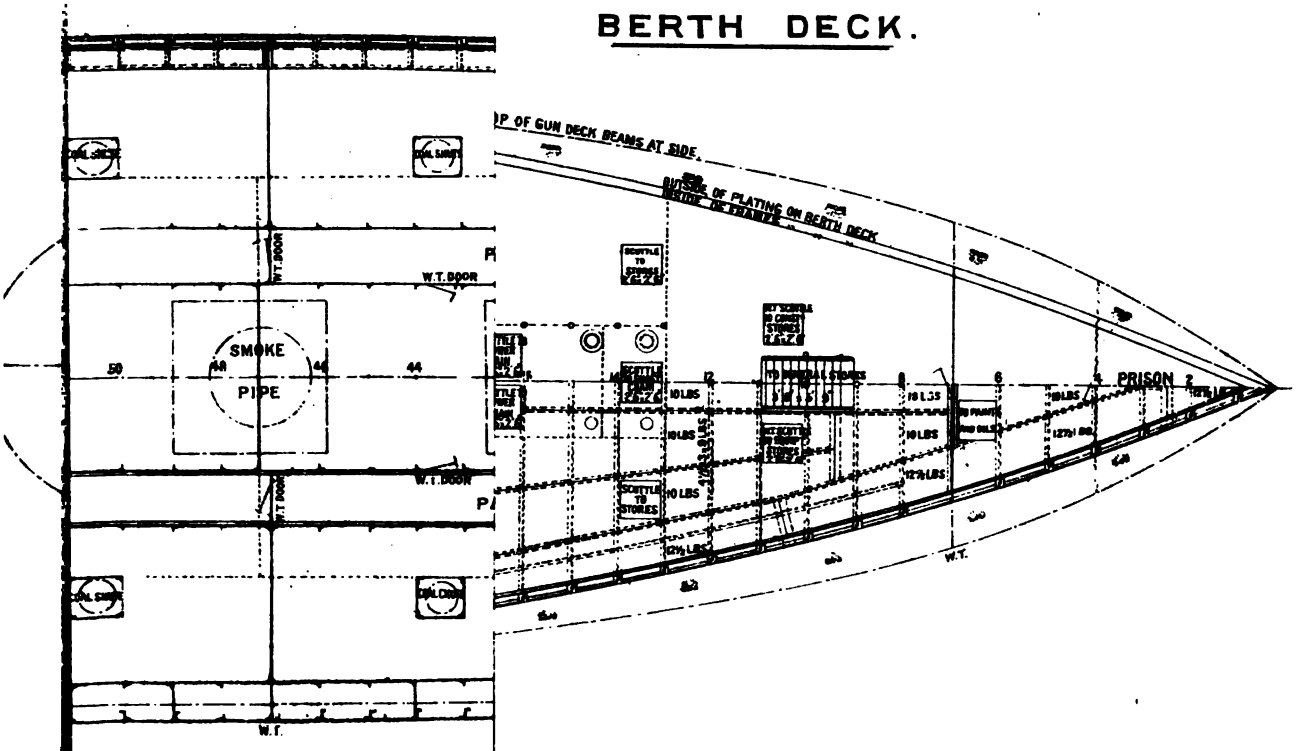
# Sheathed Protected Cruiser.

## Denver Class.

### Hold and Magazines.




### Berth Deck.

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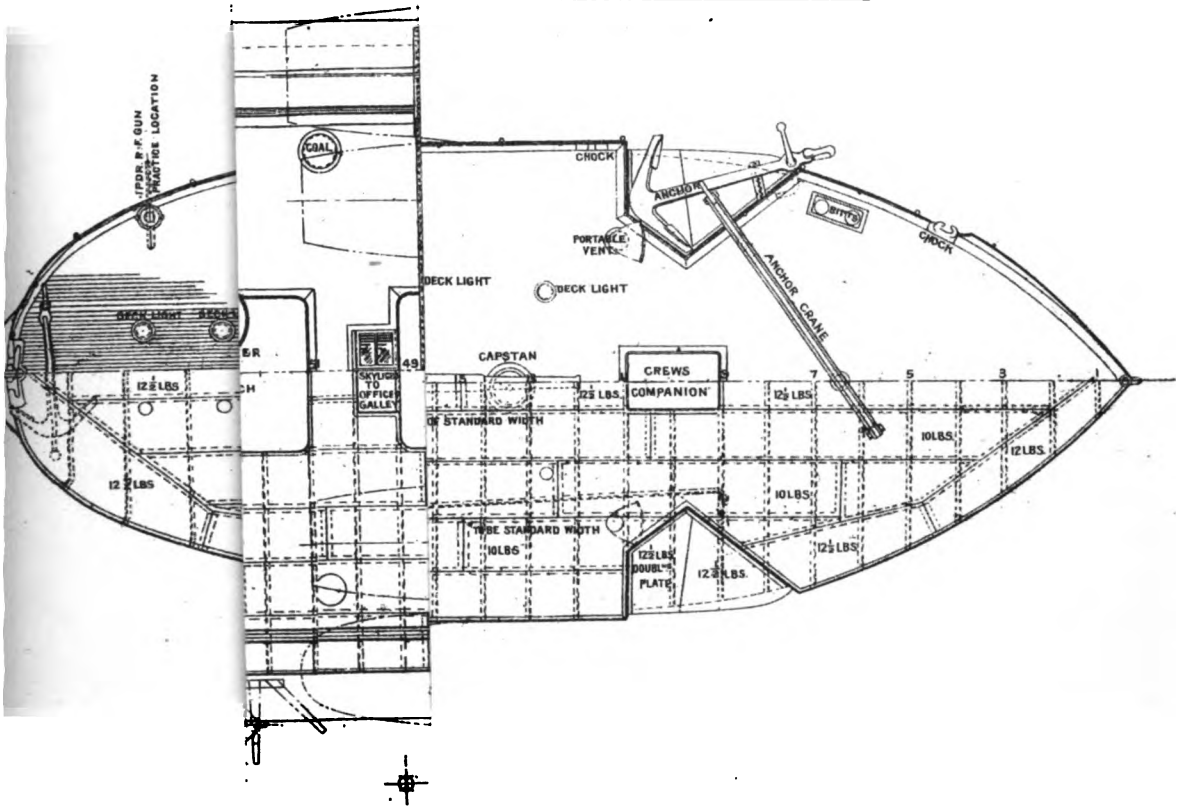
# SHEATHED PROTECTED CRUISER
## DENVER CLASS
### TACOMA, GALVESTON, DES MOINES, &c.

## MAIN DECK.




## GUN DECK.

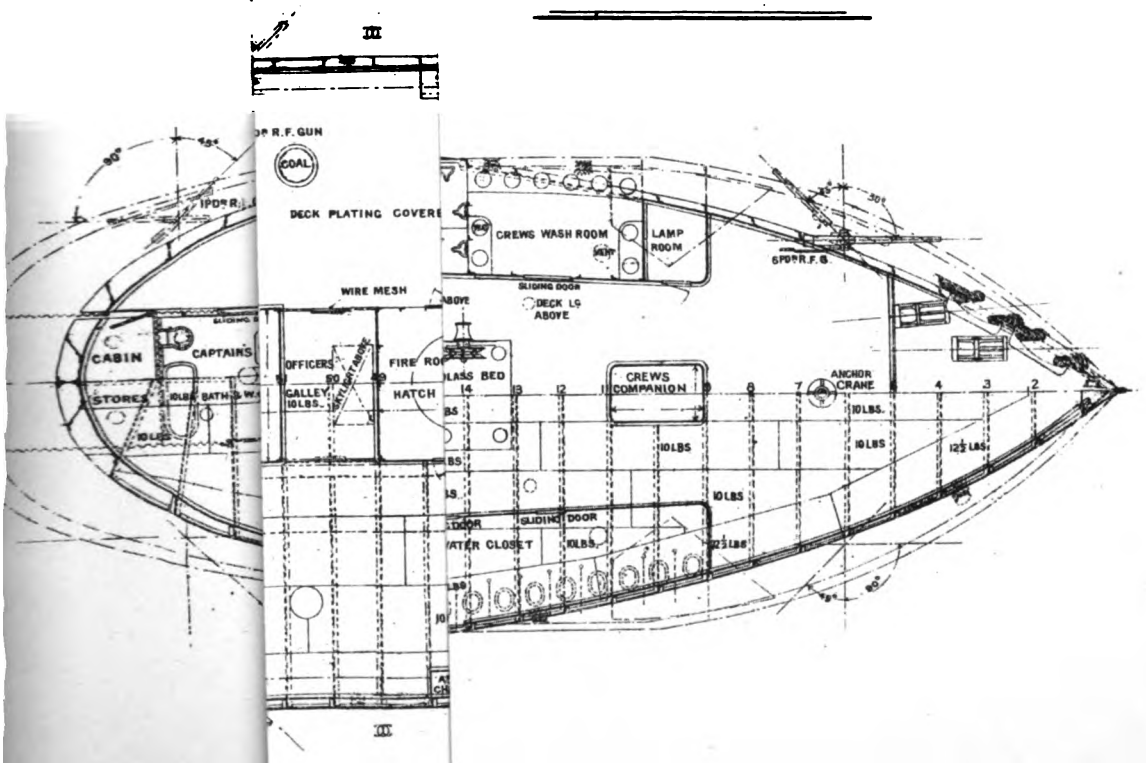



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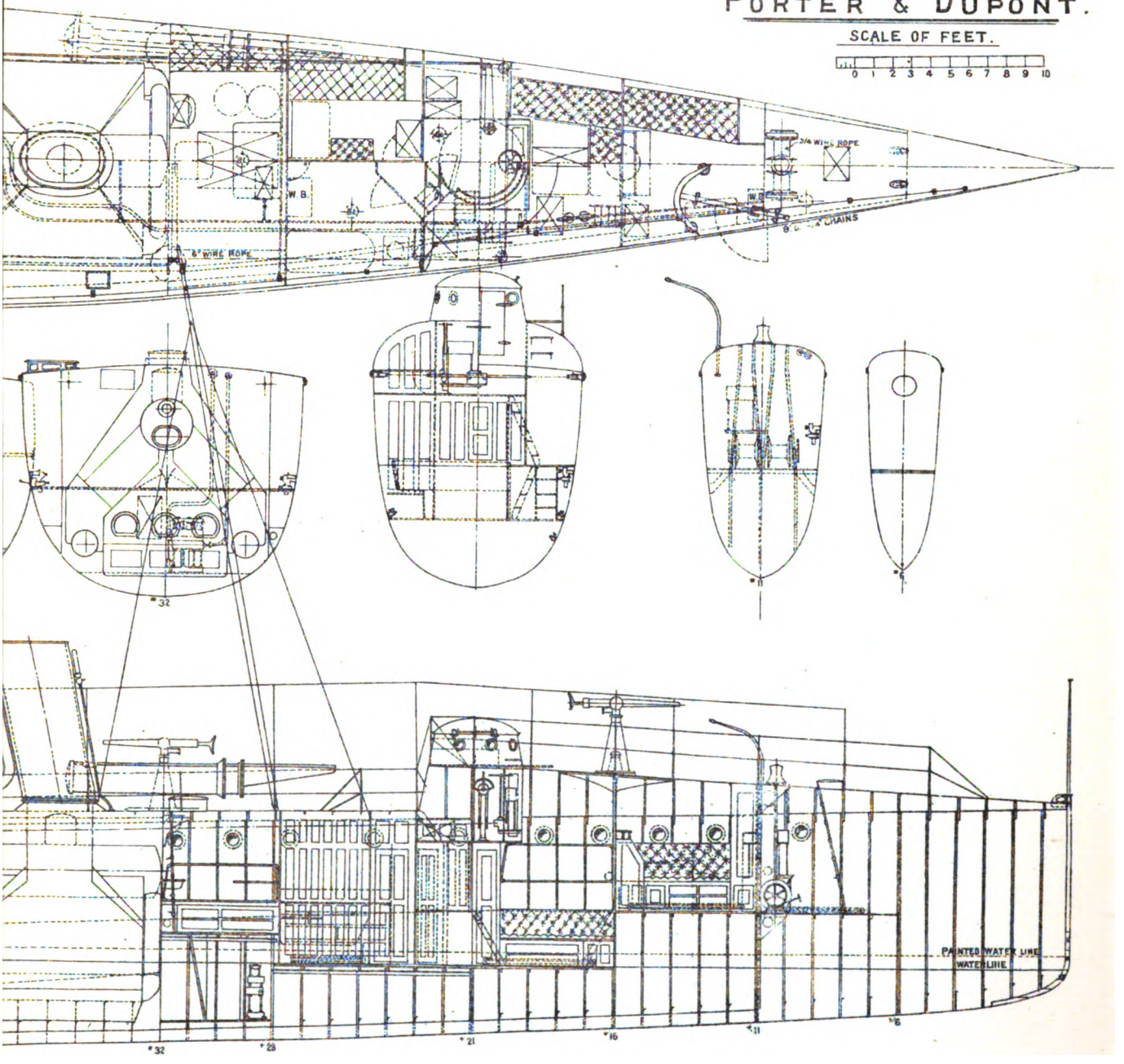
Plate XXIII.

# U. S. Torpedo Boats.
# Porter & Dupont.

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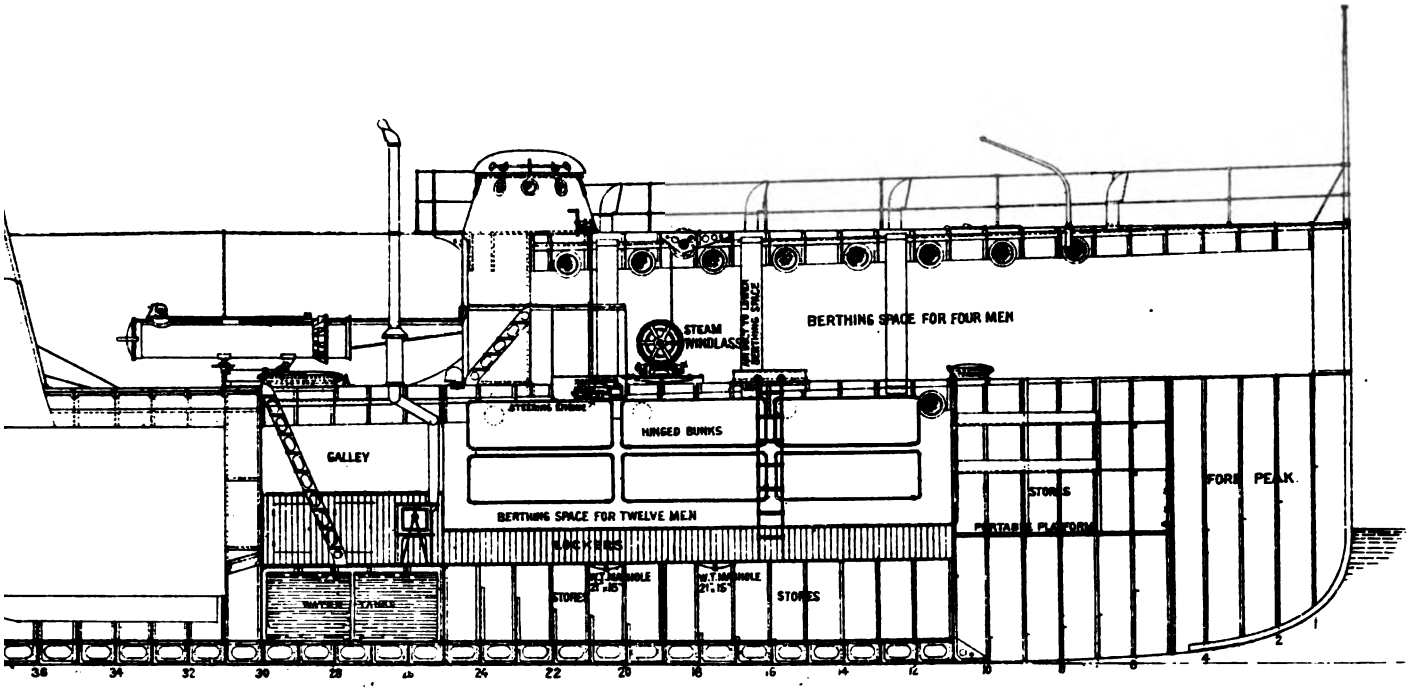
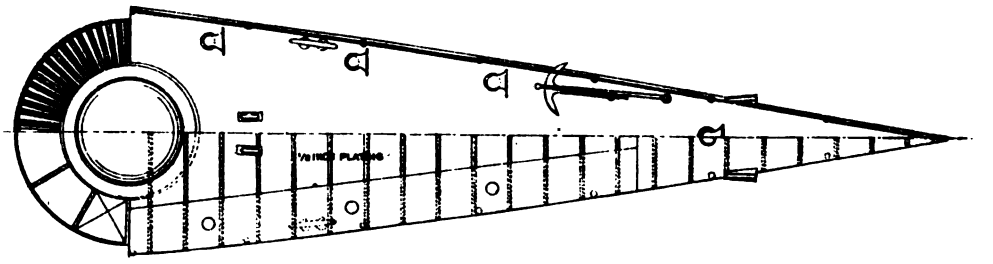
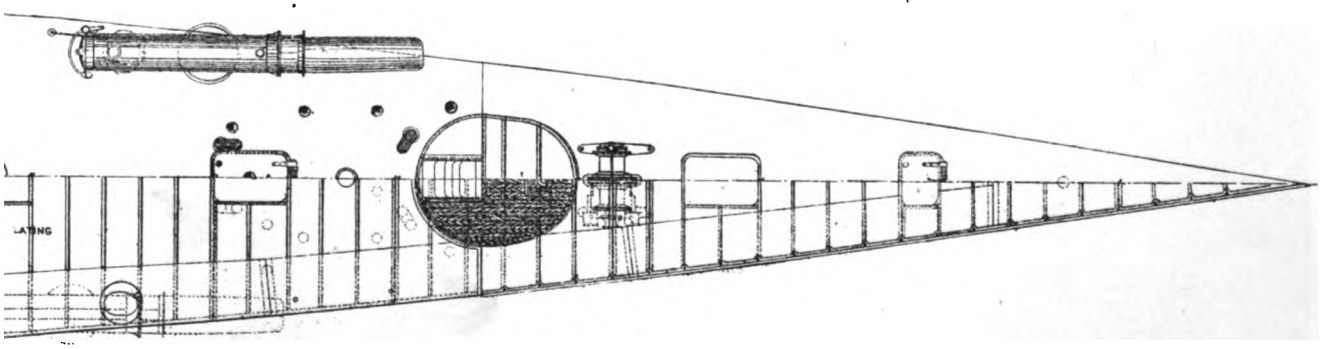
# U. S. Torpedo Boat.
## Rowan.

Scale 1/8 Inch = 1 Foot.





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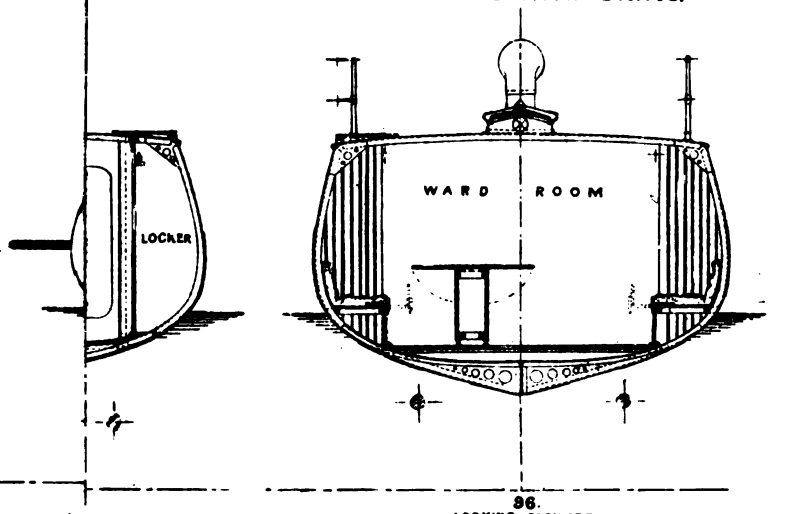
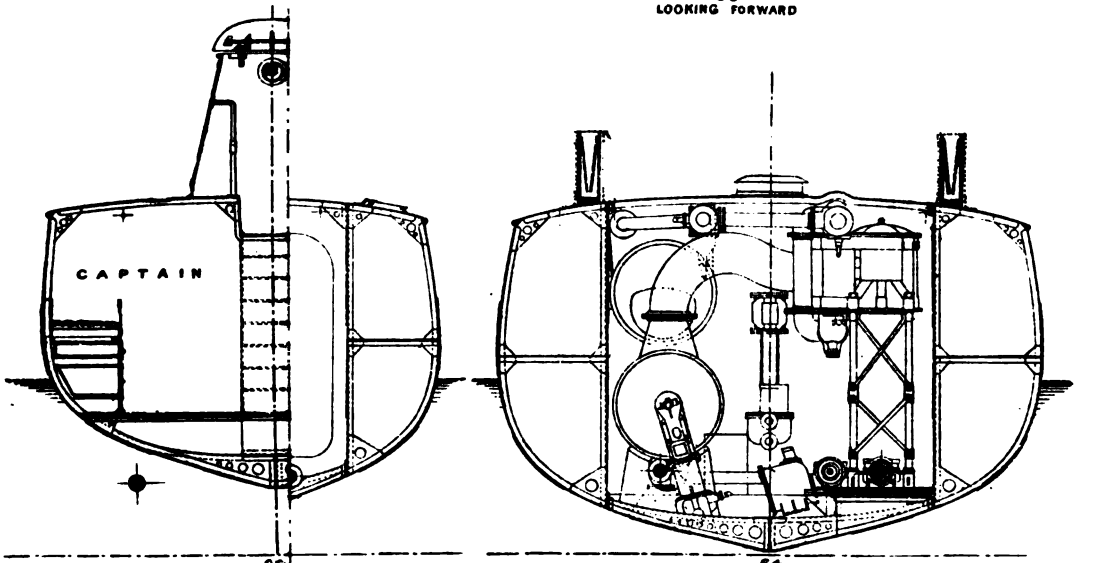
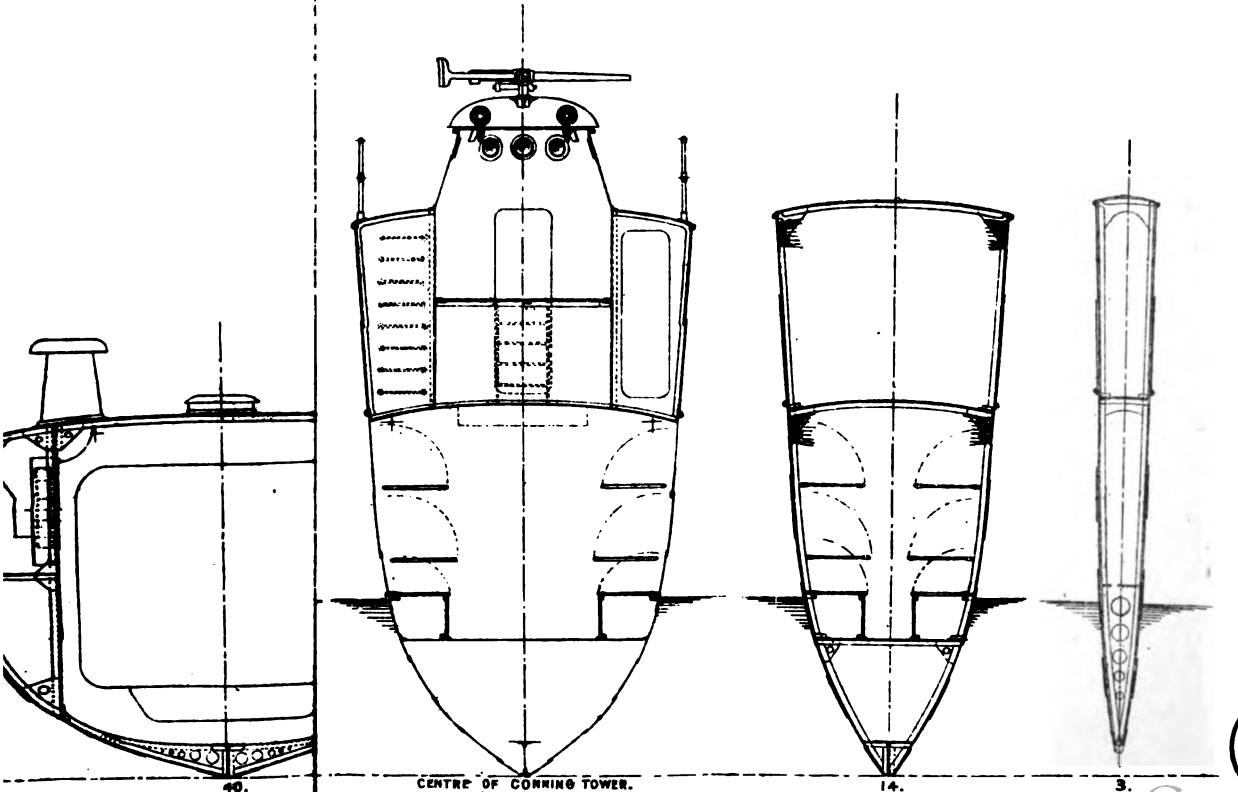



36.
LOOKING FORWARD




69. 64.



40. CENTRE OF CONNING TOWER. LOOKING AFT. 14. 3.

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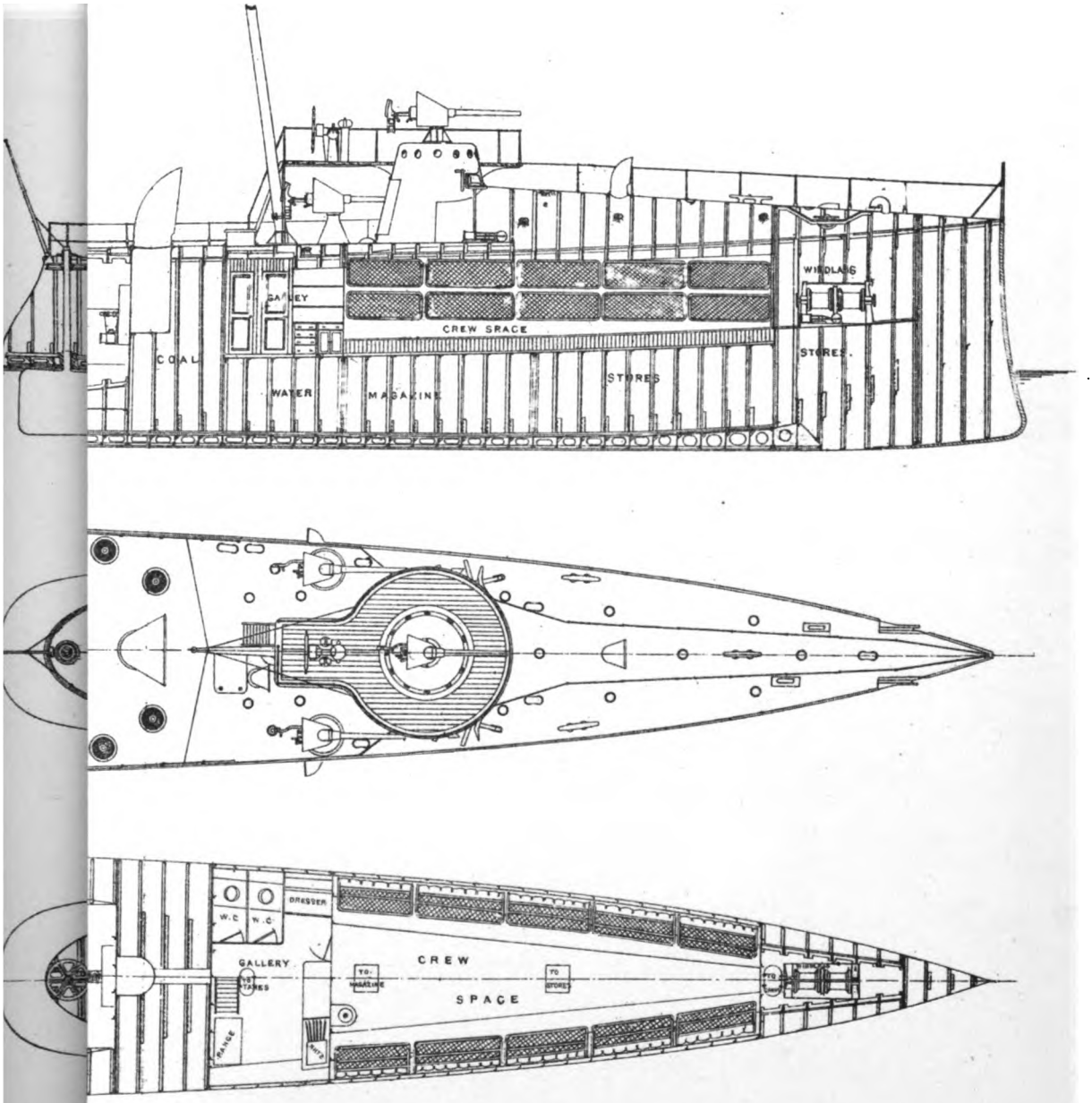
United States.

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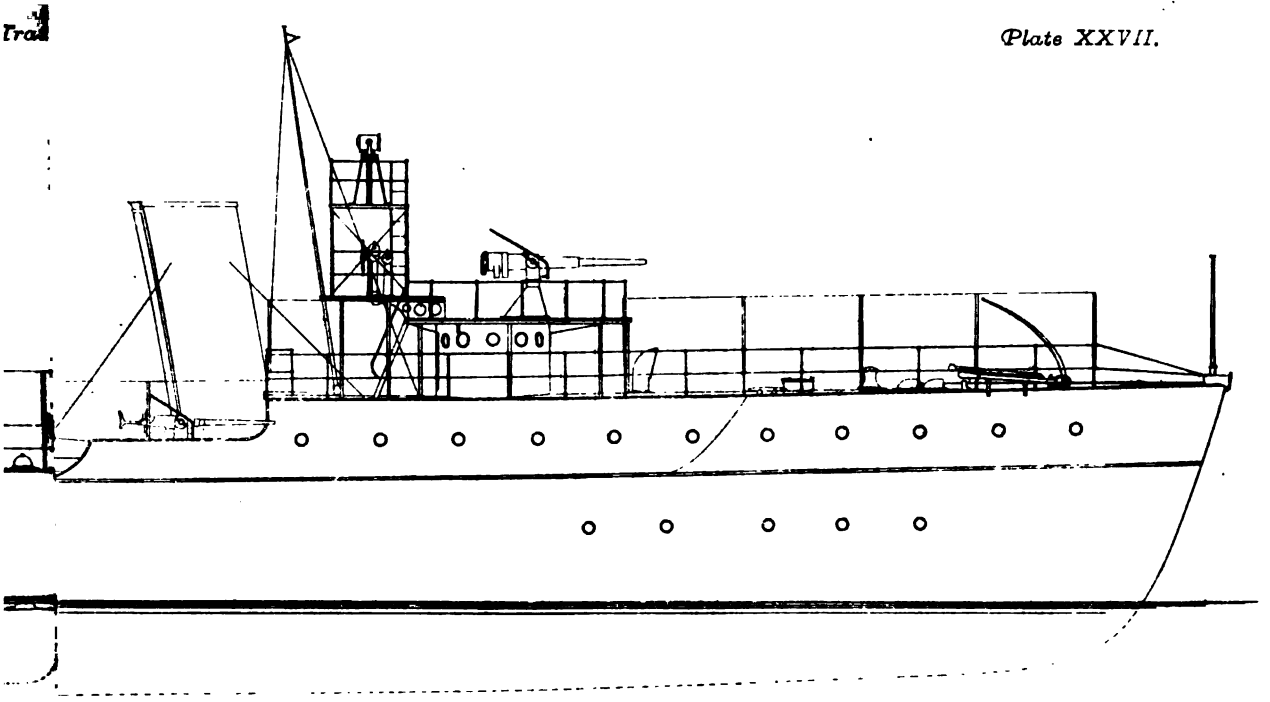

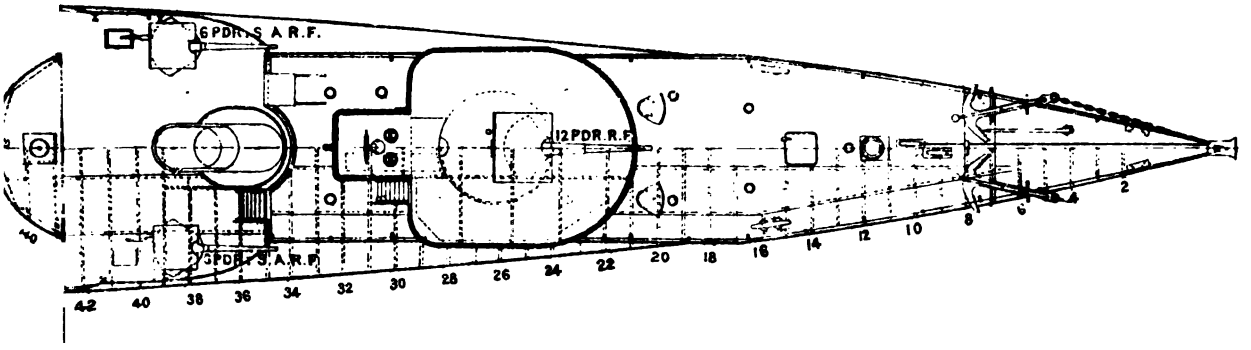


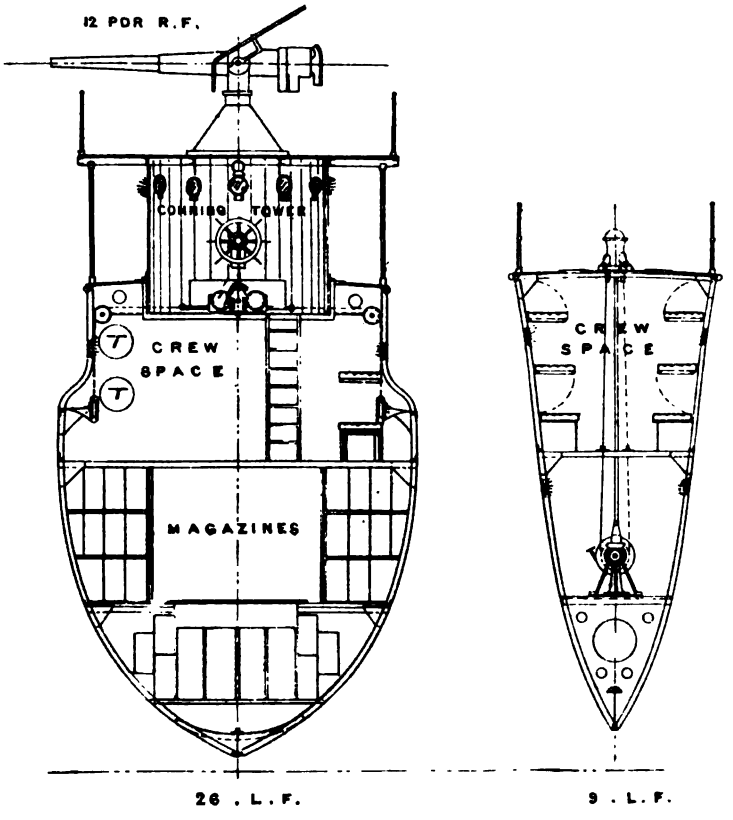


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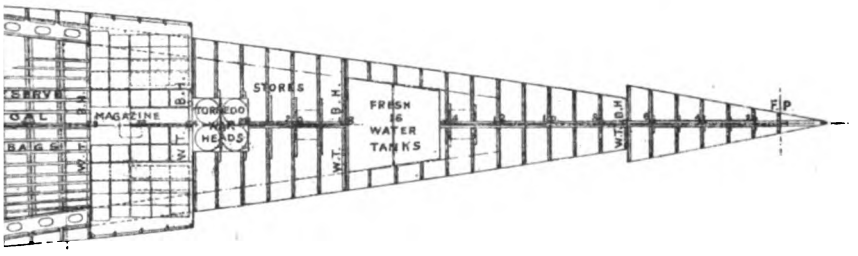


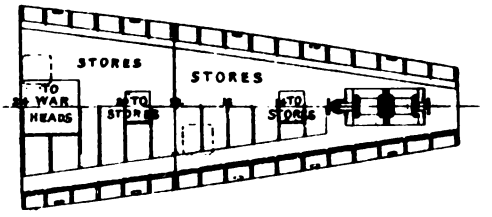


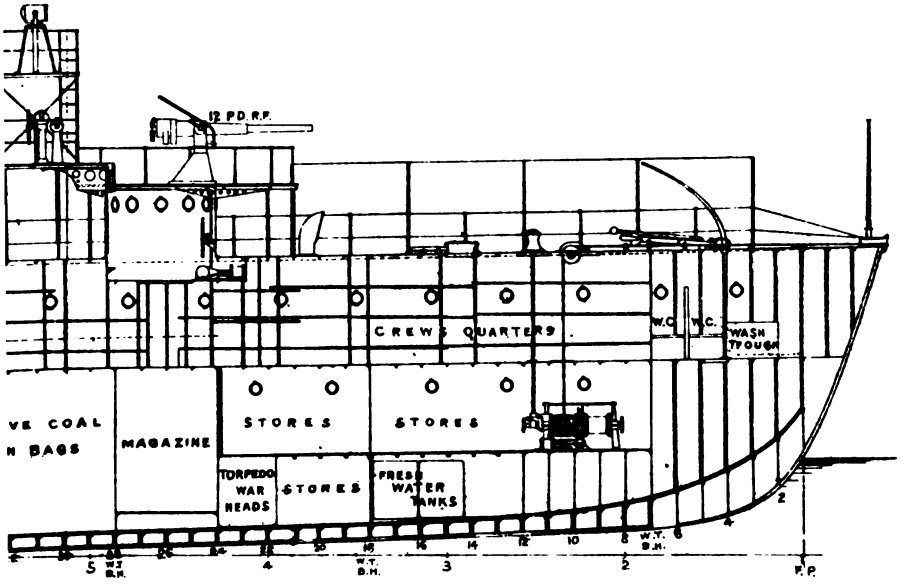


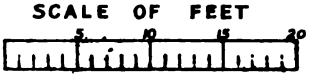


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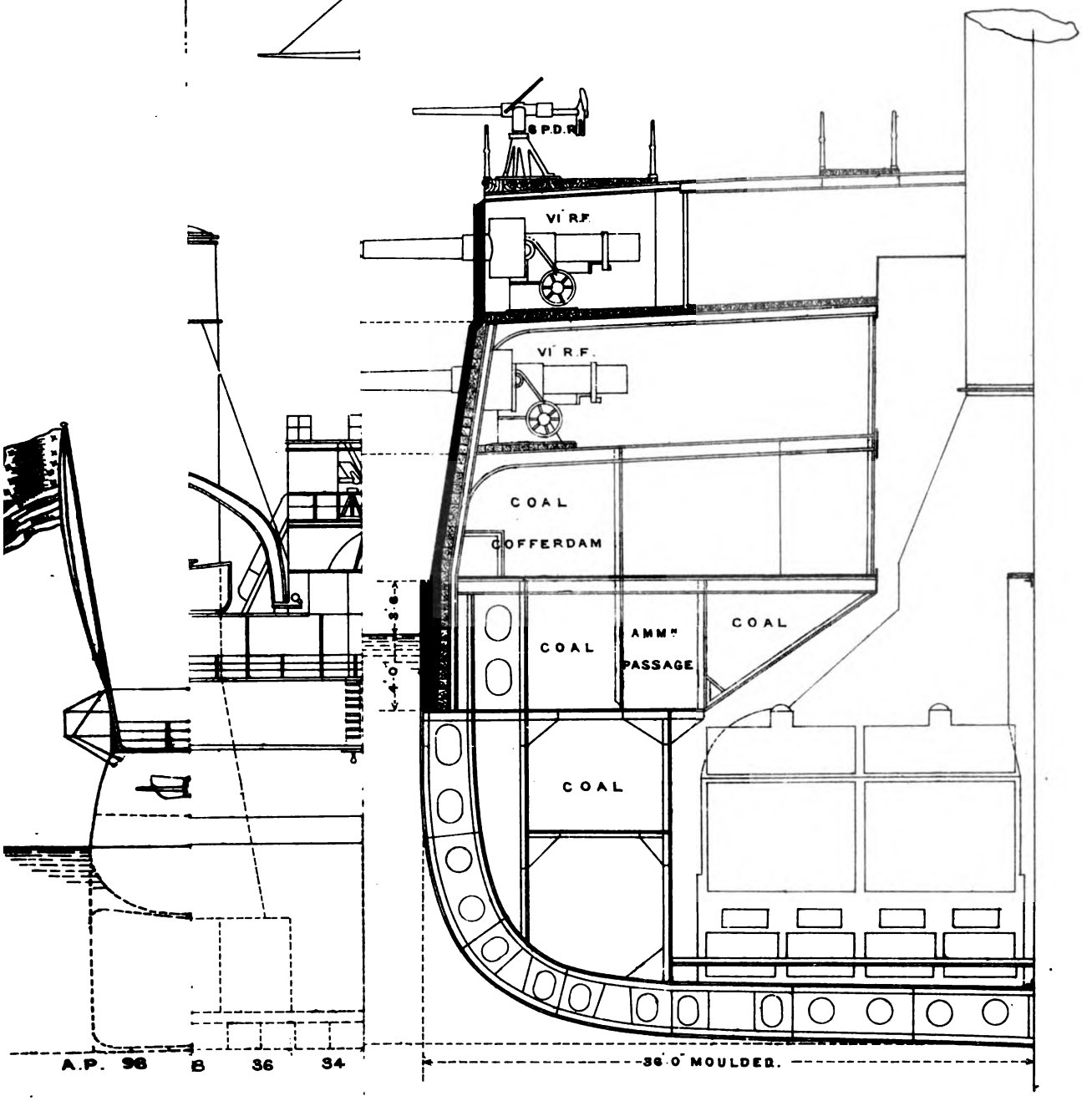


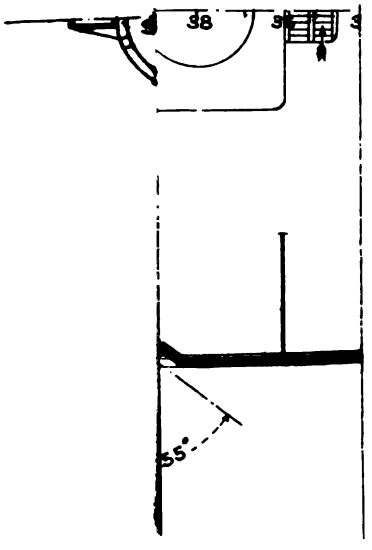


U.S. BATTLE SHIPS.
"MAINE" "MISSOURI" "OHIO"
MAIN DECK.

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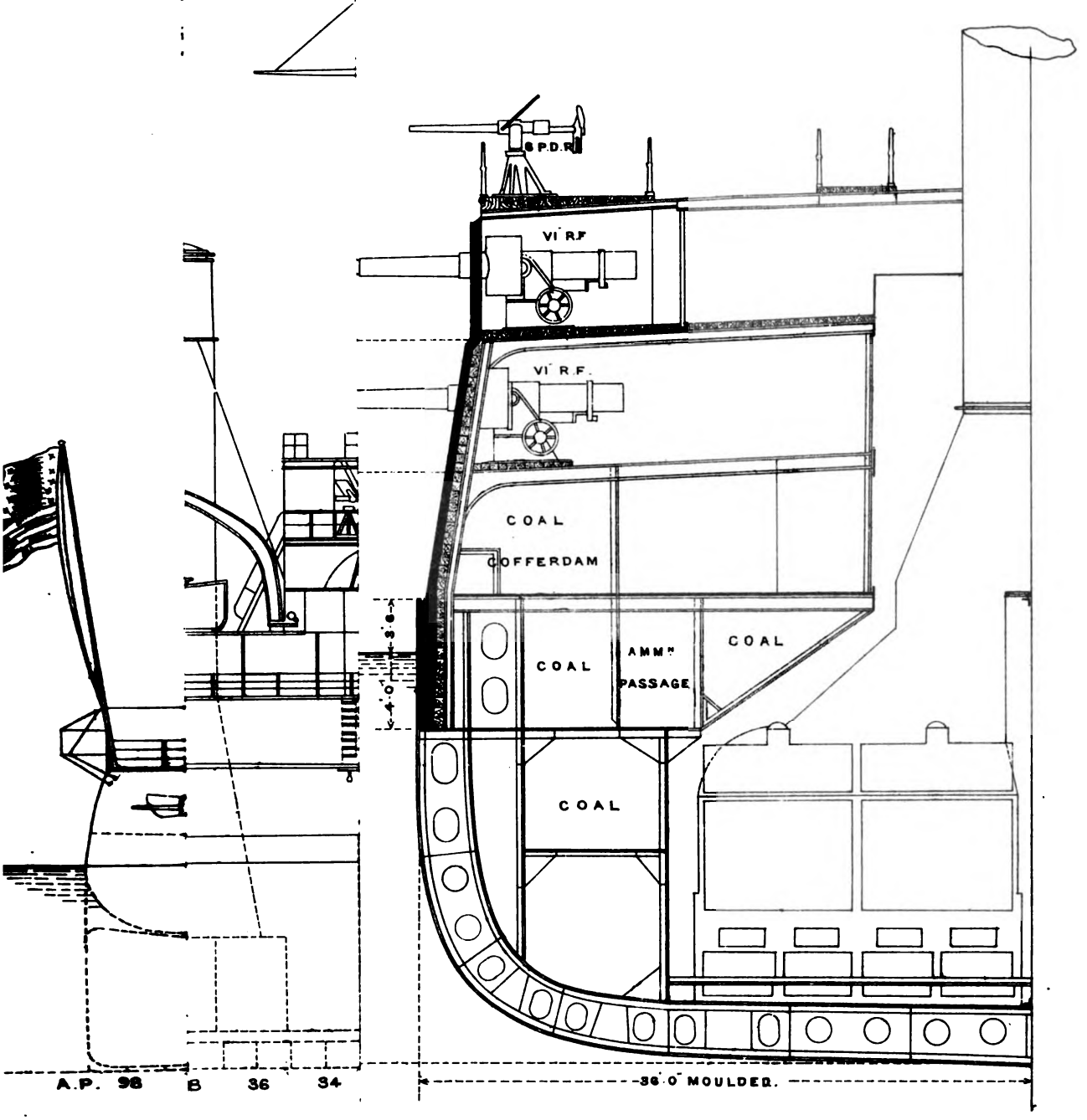


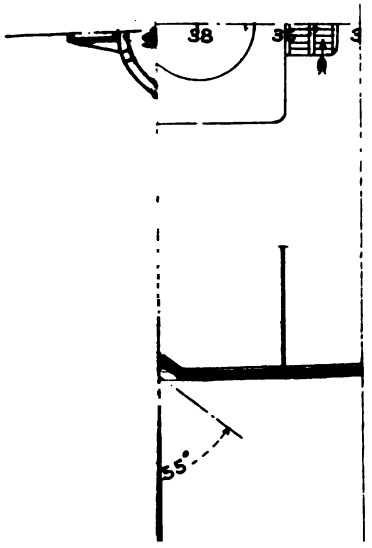


U.S. BATTLE SHIPS.
"MAINE" "MISSOURI" "OHIO"
MAIN DECK.

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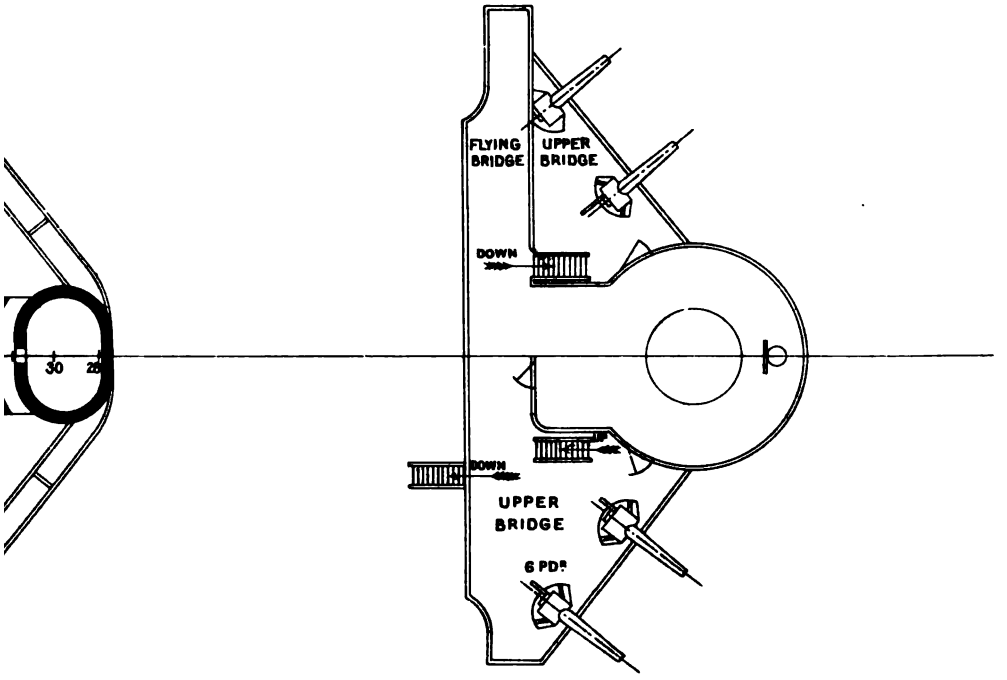
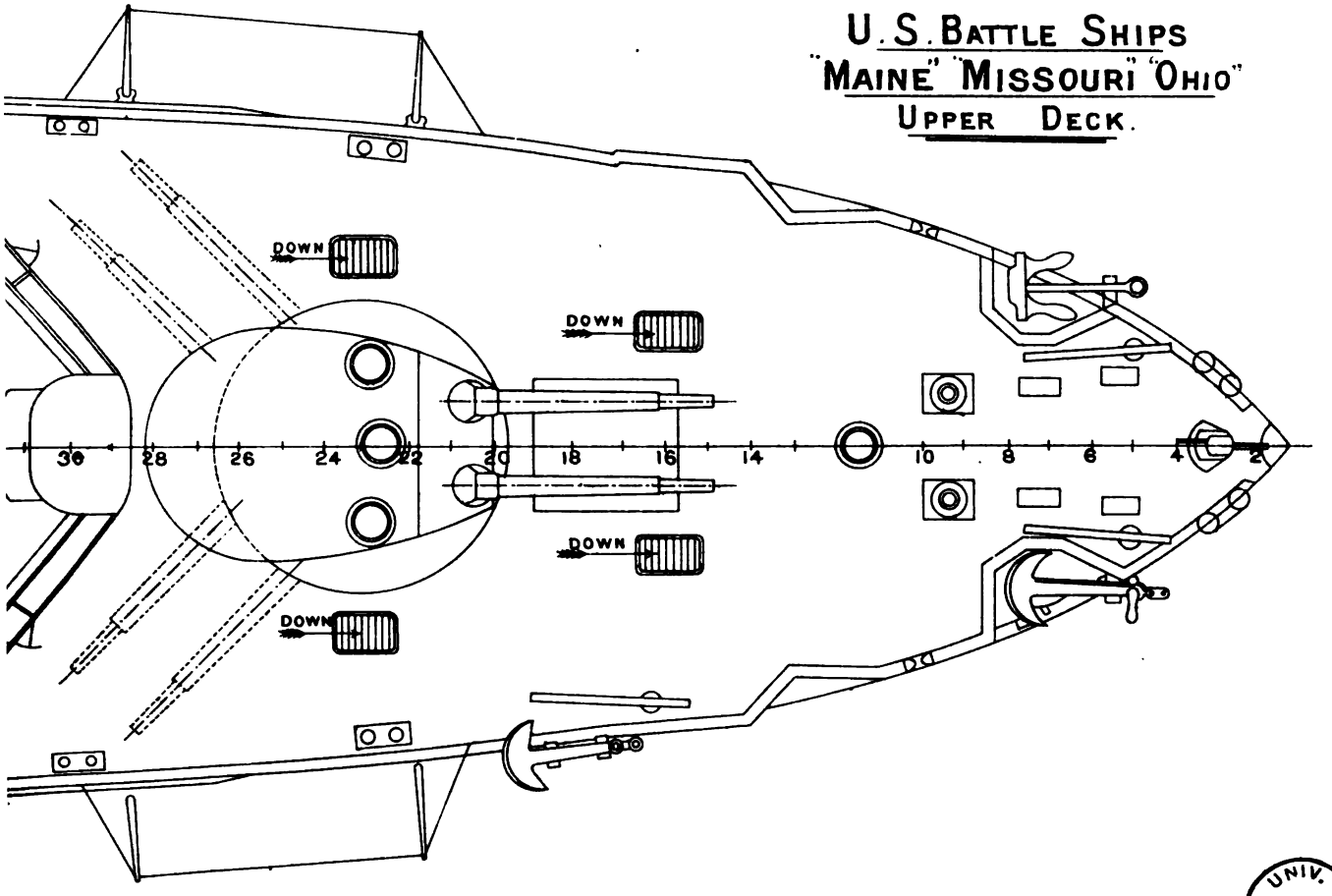
# U.S. Battle Ships
## "Maine" "Missouri" "Ohio".
### Bridges and Stowage of Boats




# U.S. Battle Ships
## "Maine" "Missouri" "Ohio"
### Upper Deck.

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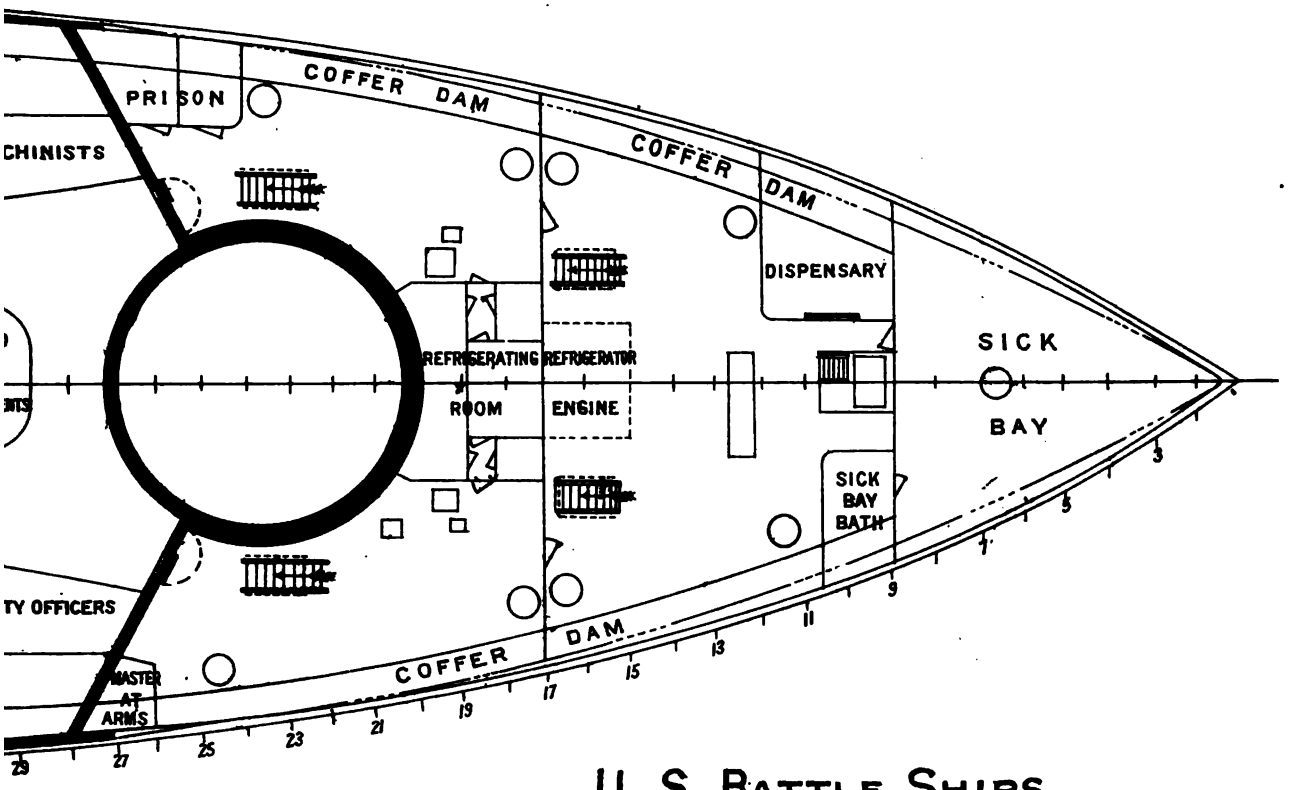
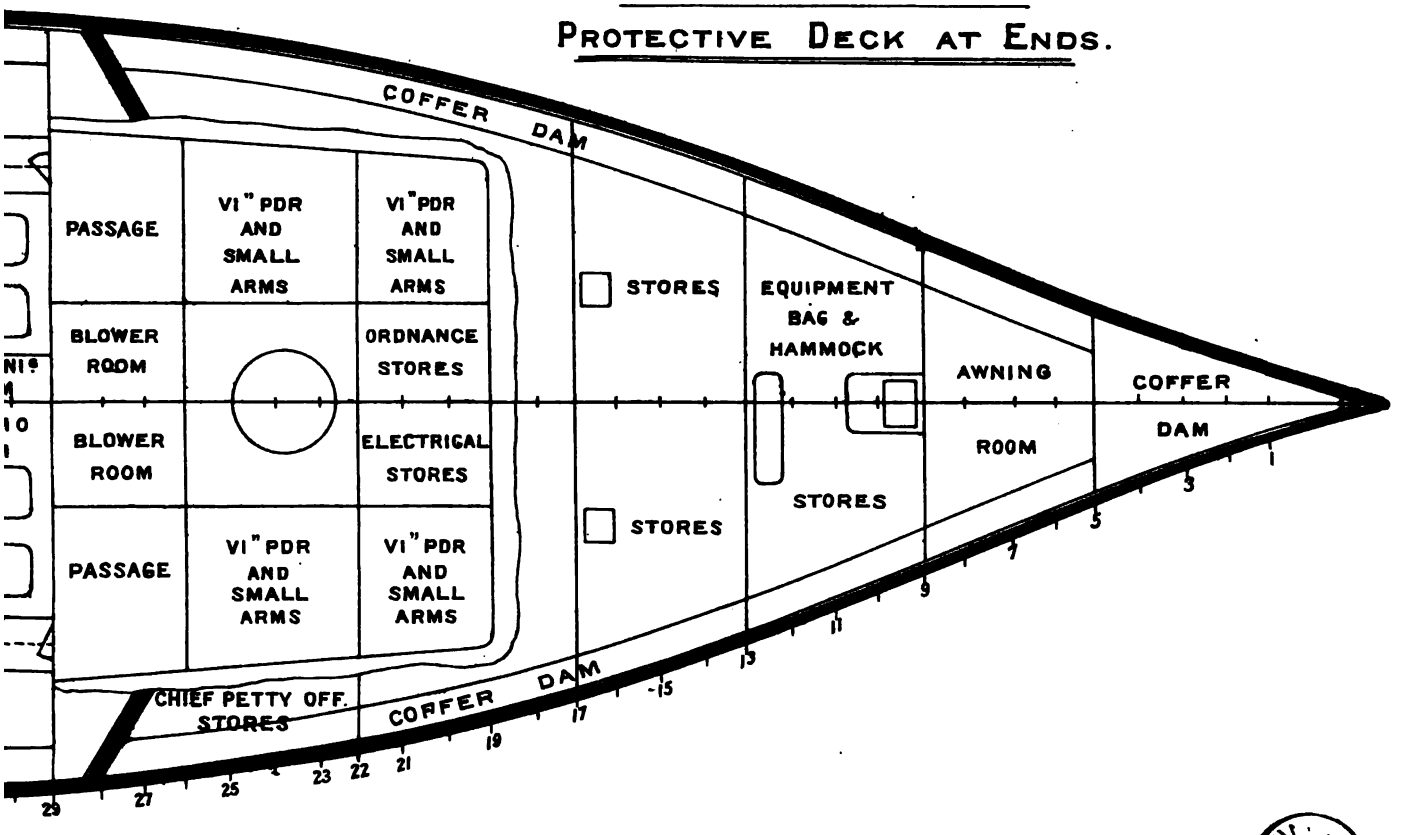
# U. S. Battle Ships.
## "Maine" "Missouri" "Ohio"
## Berth & Protective Decks.




# U. S. Battle Ships
## "Maine" "Missouri" "Ohio"
## Splinter Deck.
## Protective Deck at Ends.

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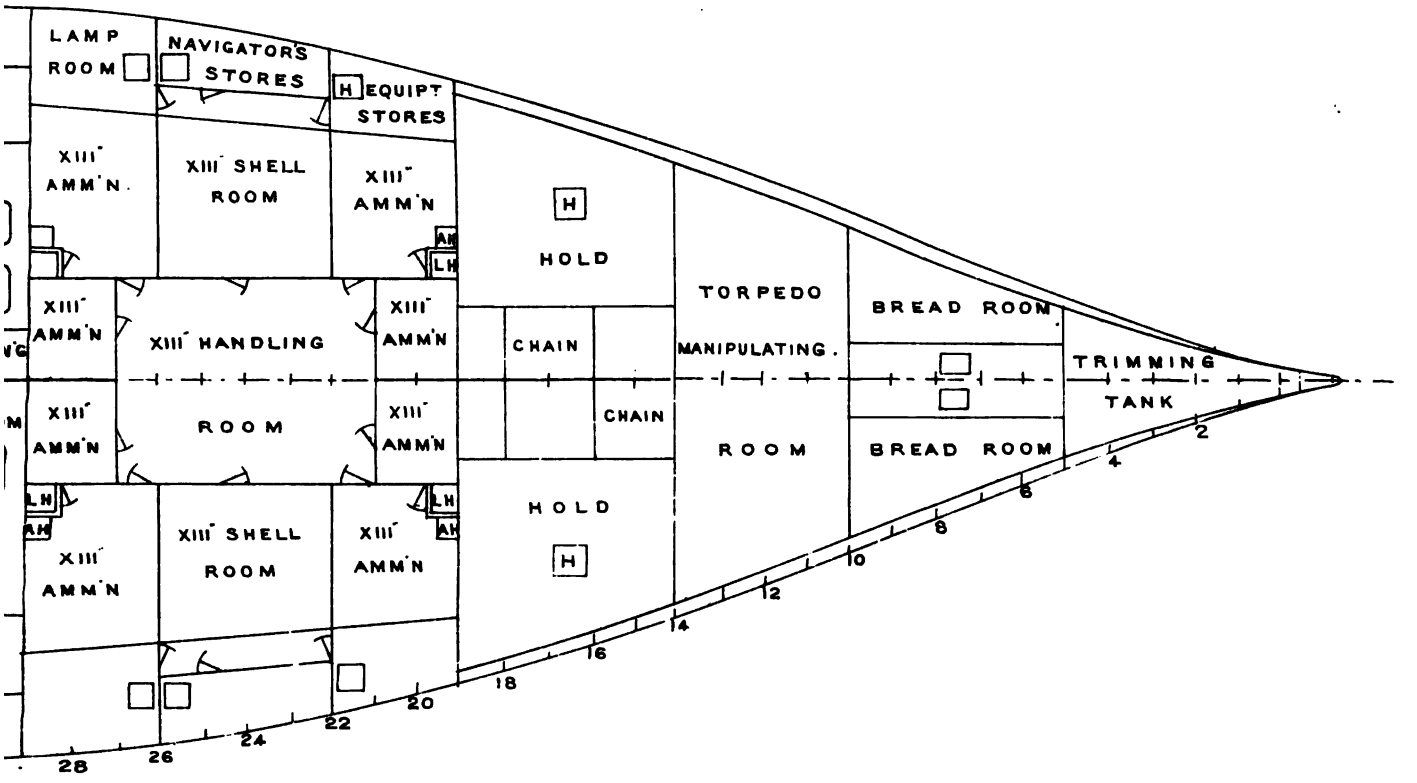
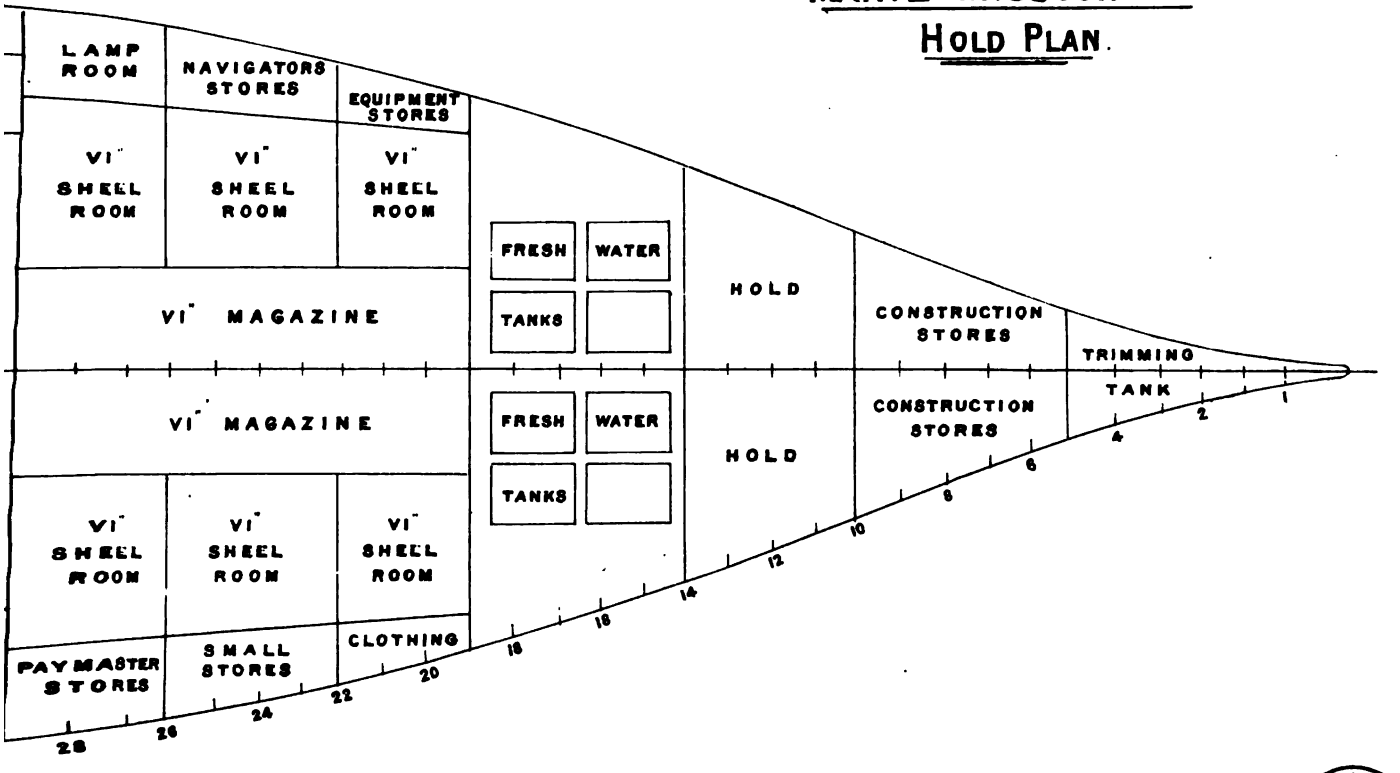
# U. S. Battle Ships "Maine" "Missouri" "Ohio"

## Platform Deck




# U. S. Battle Ships. "Maine" "Missouri" "Ohio"

## Hold Plan.

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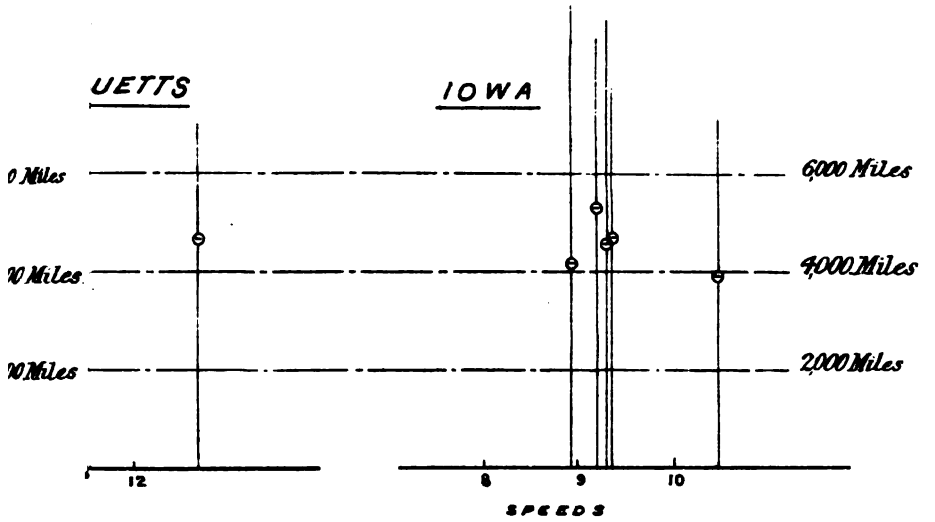
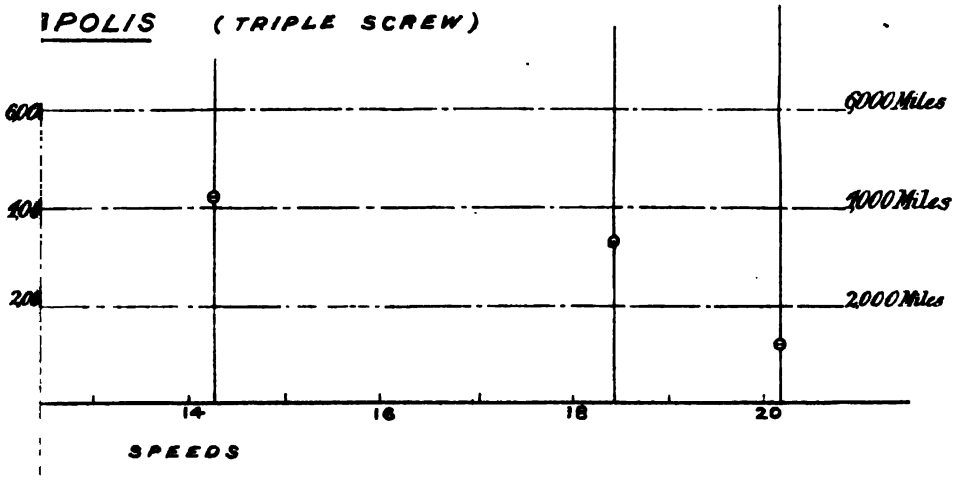
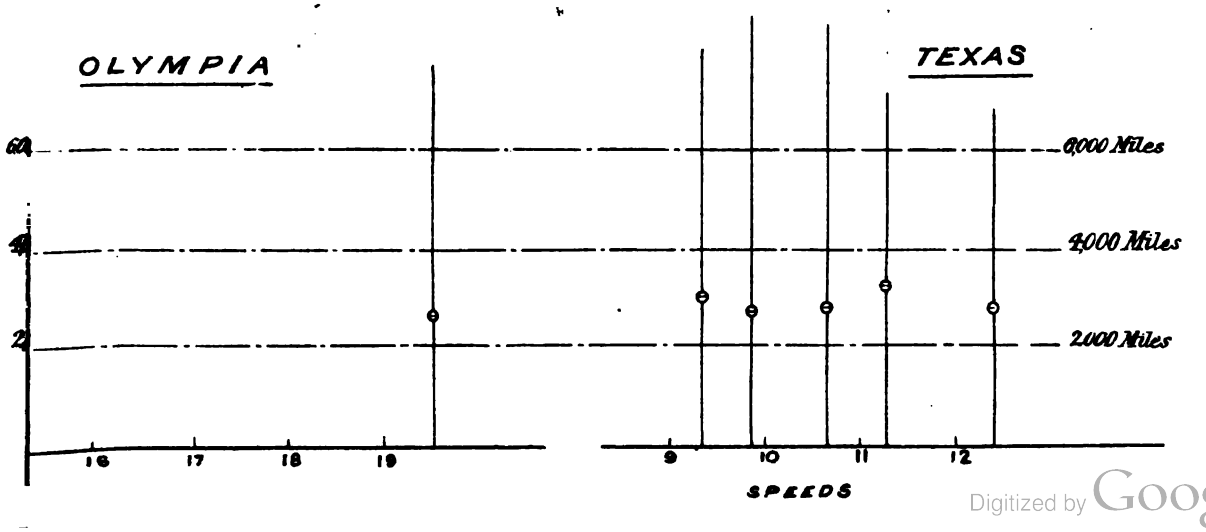
## SHIPS

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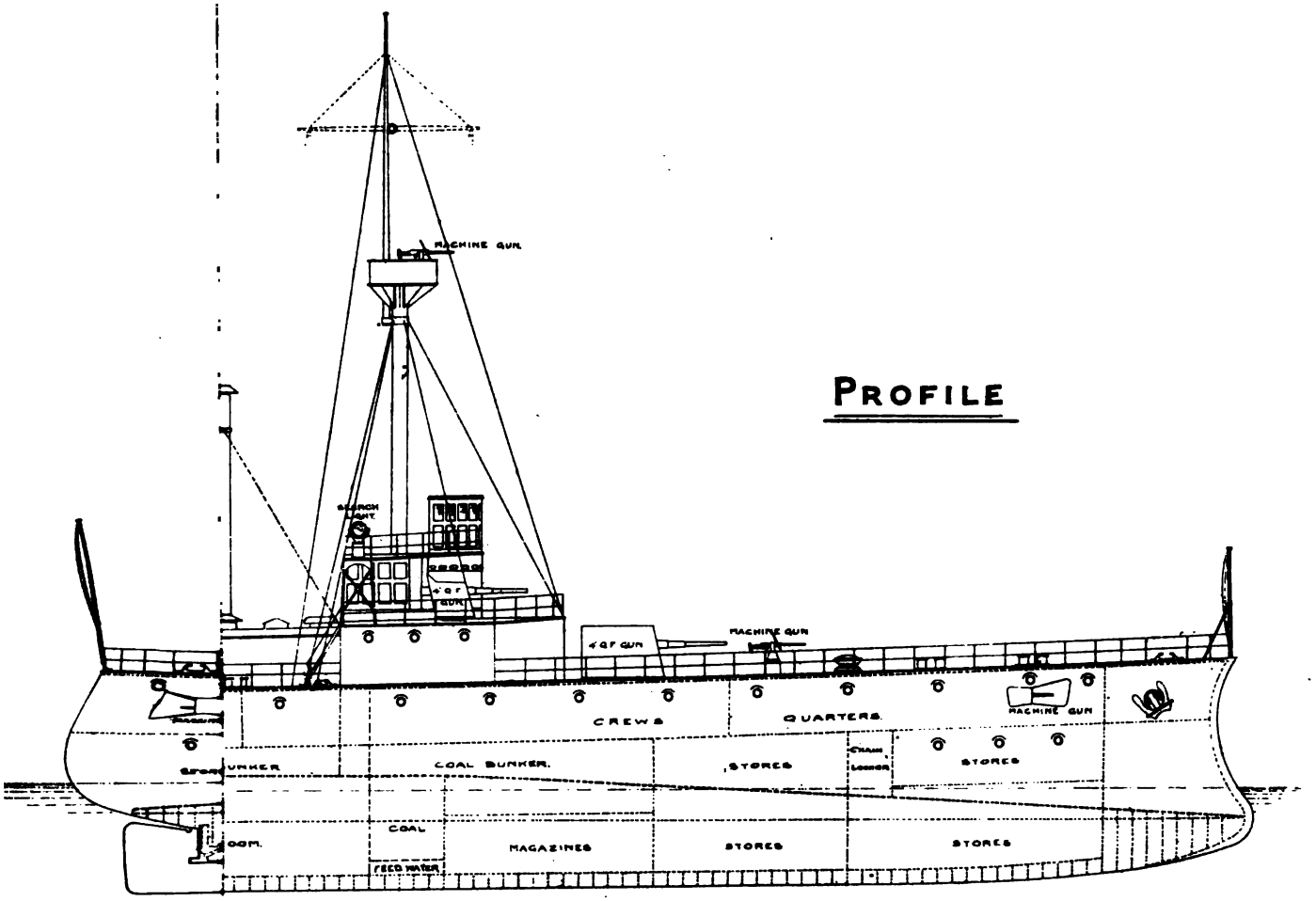
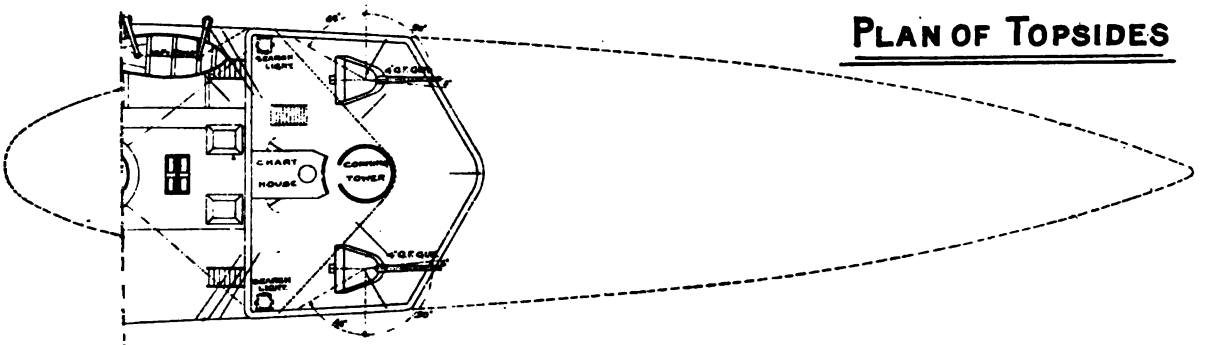
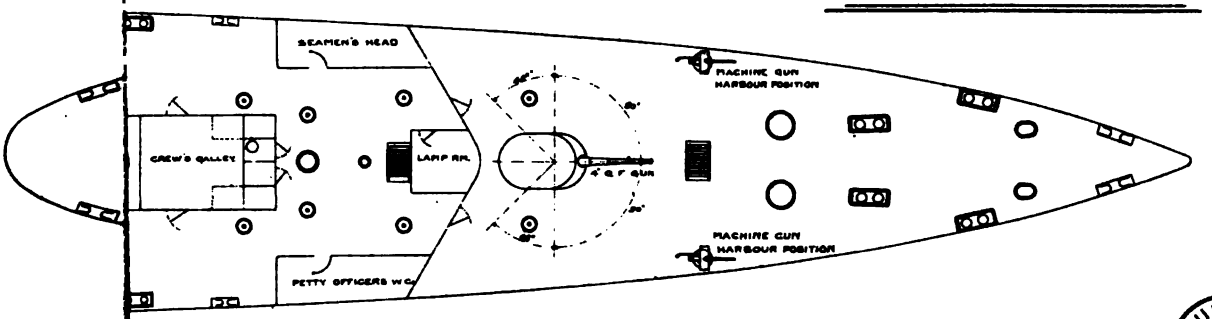
Scout.




PROFILE




PLAN OF TOPSIDES




PLAN OF UPPER DECK

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FIG 3.

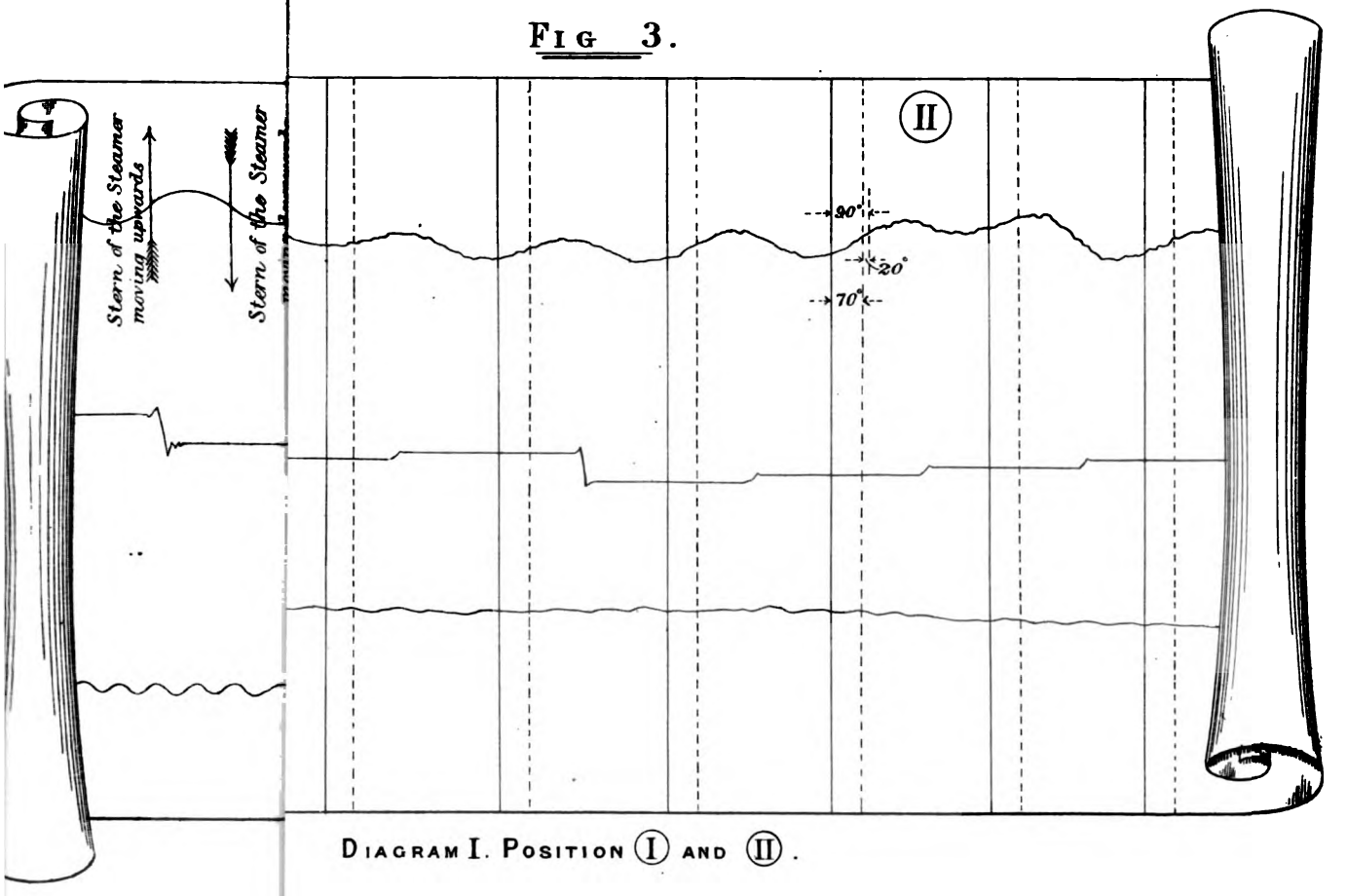
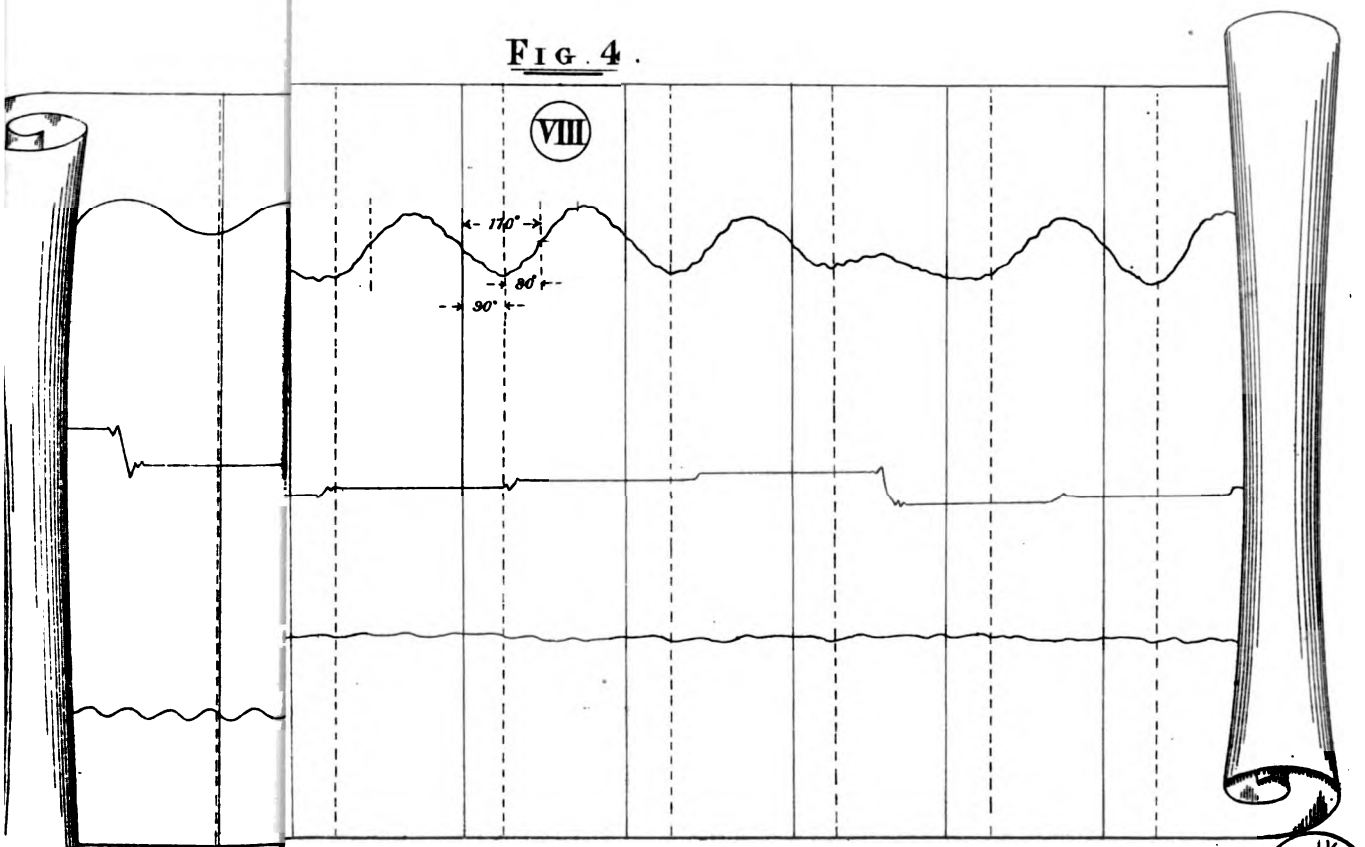



DIAGRAM I. POSITION (I) AND (II).

FIG. 4.




b

DIAGRAM II. POSITION (VIII).

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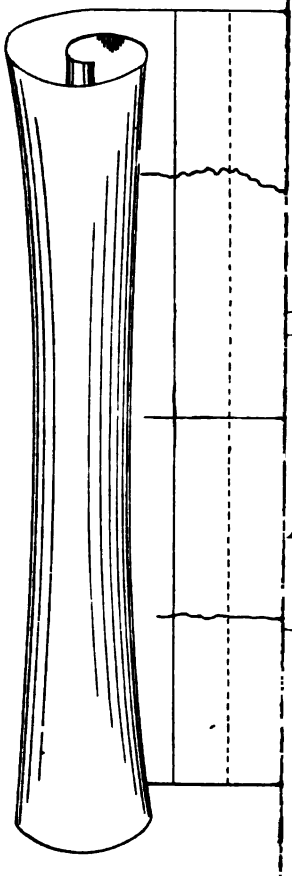


FIG. 11.

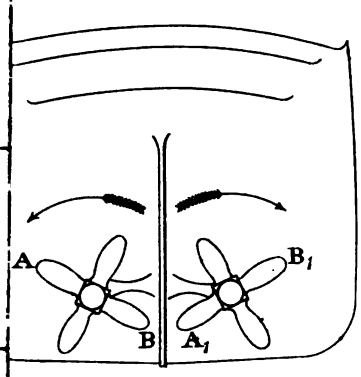



FIG. 16.

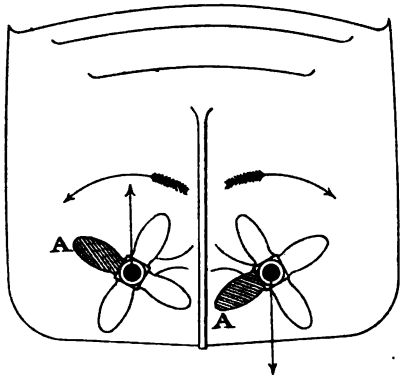



FIG. 12.

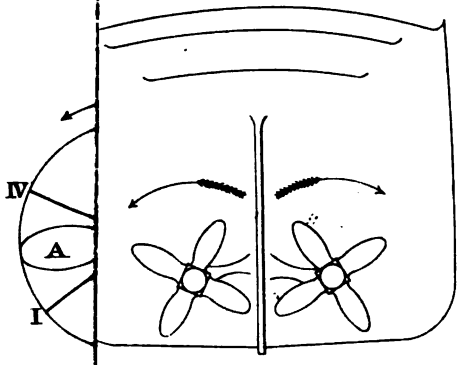



FIG. 15.

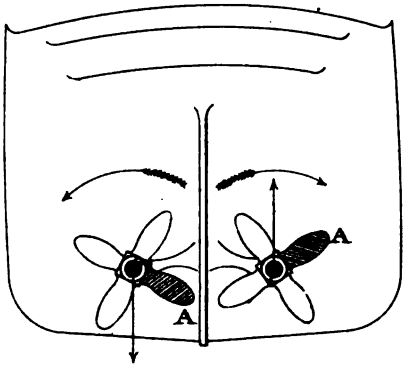



FIG. 17.

FIG. 13.

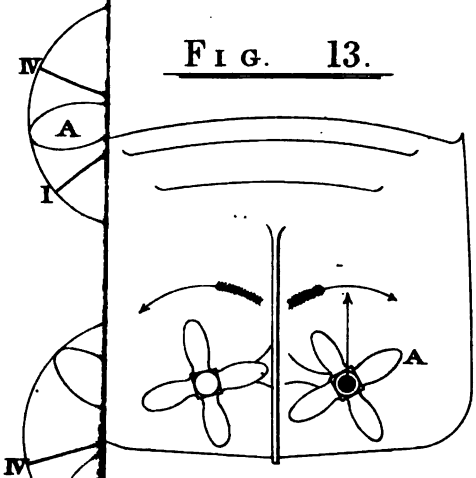
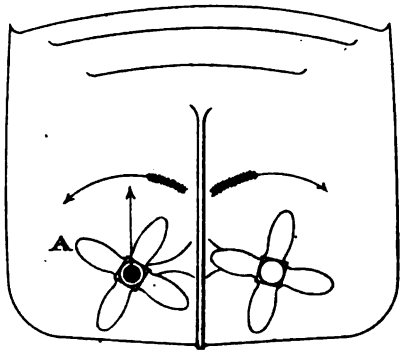



FIG. 14.

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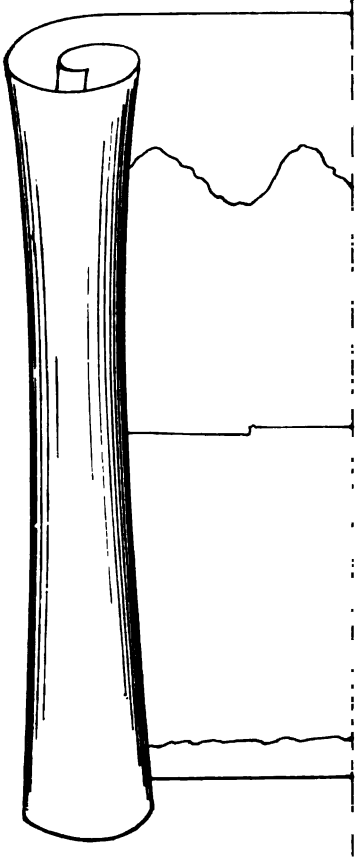

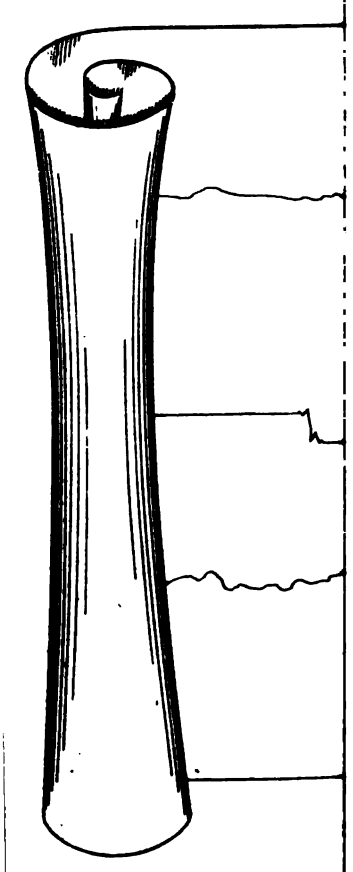

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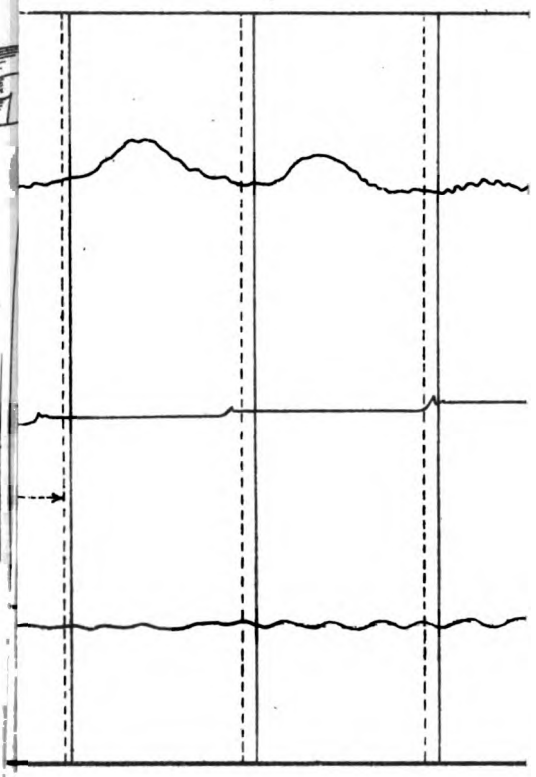

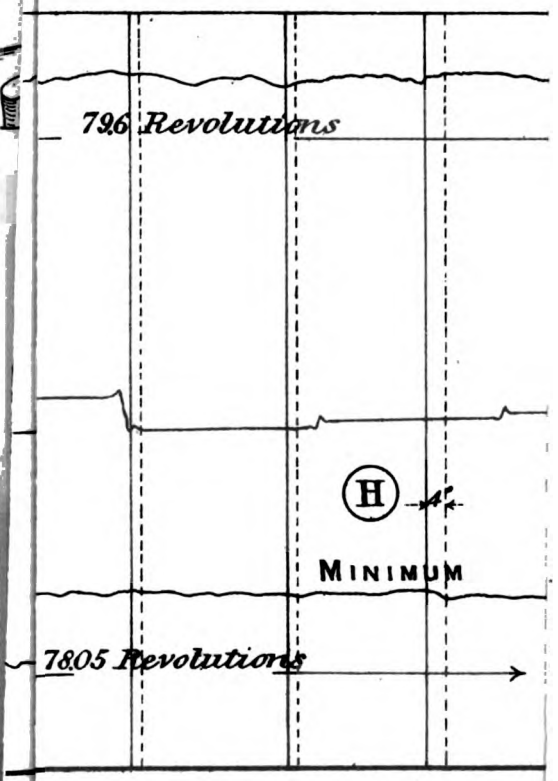


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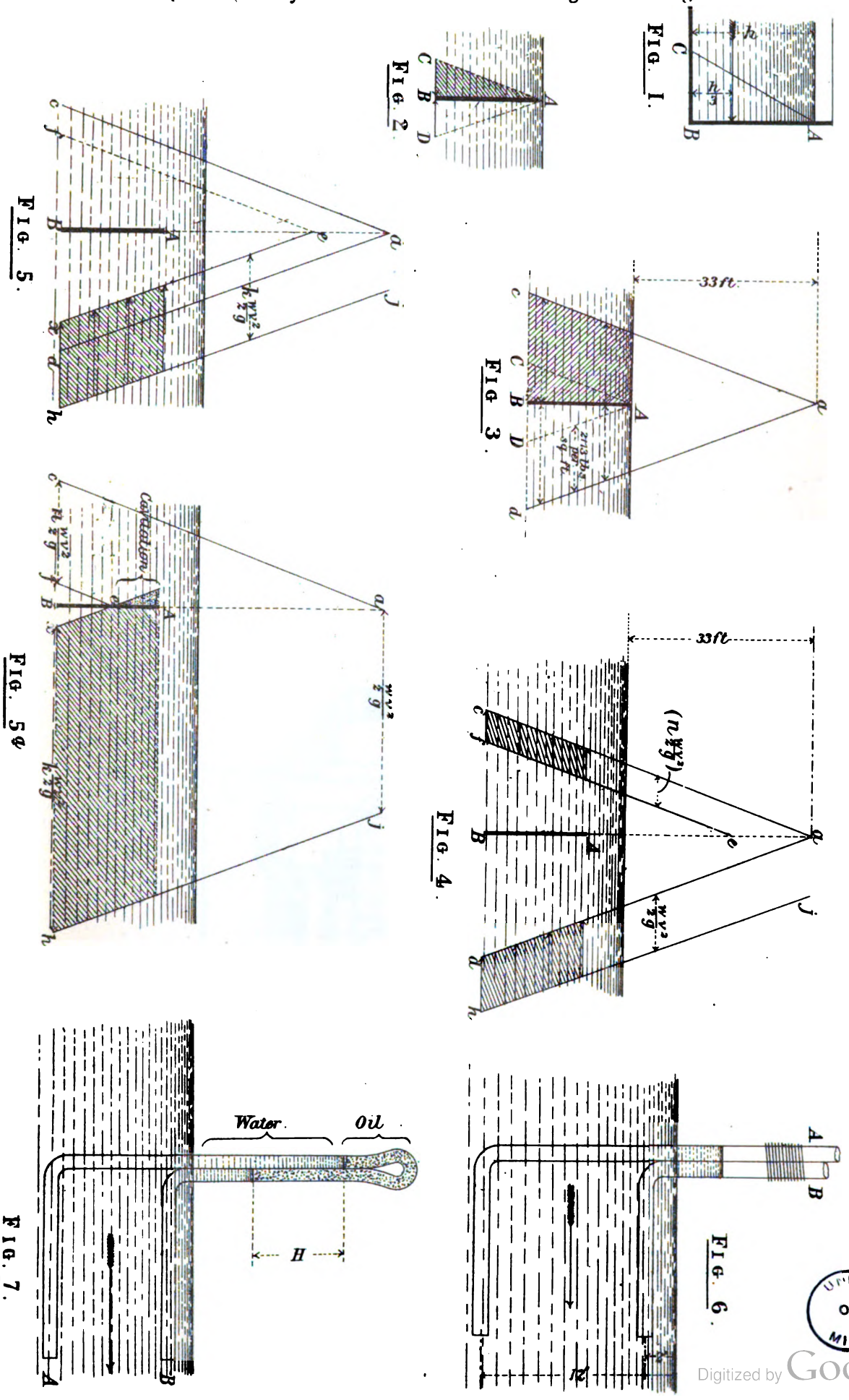
*To Illustrate Mr. C. Humphrey Wingfield's Paper: The Influence of Depth of Immersion on the Distribution of Pressure over a submerged moving Plate.*

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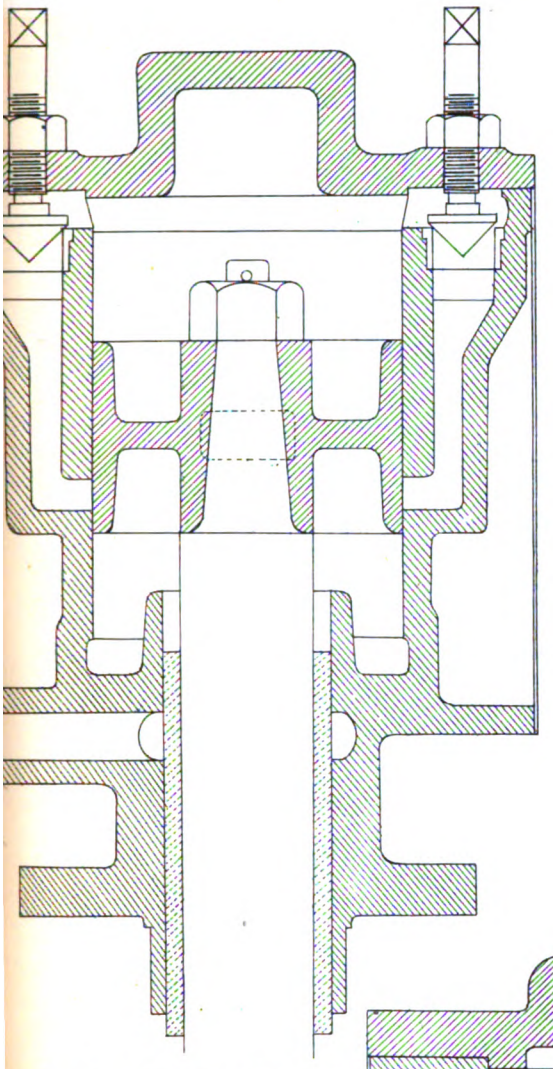
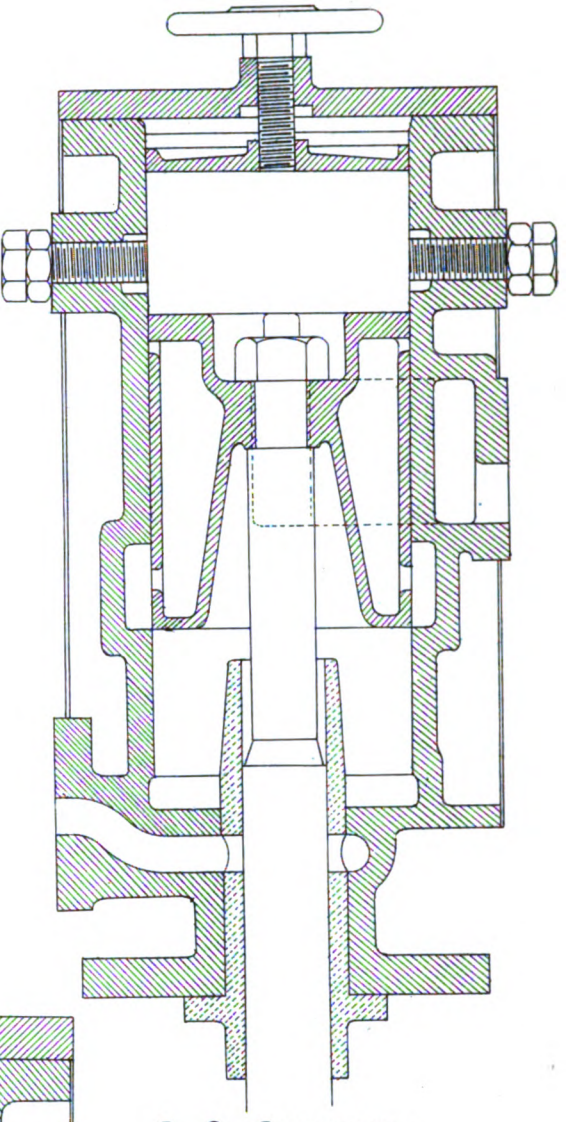
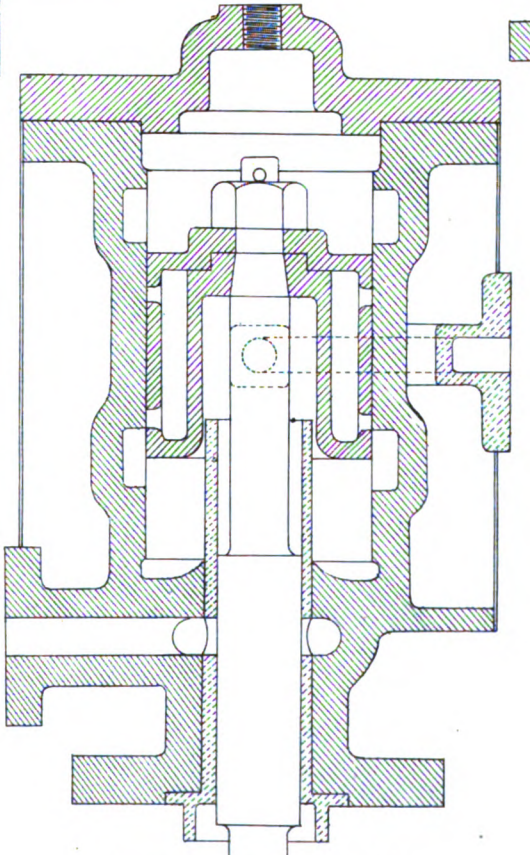
*To Illustrate Mr. David Joy's Paper on a New Assistant Cylinder.*



S. S. BITTERN.
FIG. 1.



S. S. SULTANA
FIG. 3.



ENGINES FITTED BY
MESSRS. LAIRD BROS.
FIG. 2

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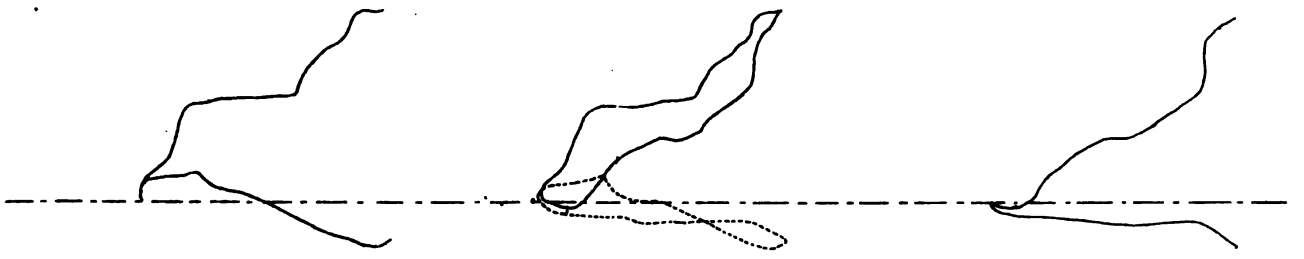
*To Illustrate Mr. David Joy's Paper on a new Assistant Cylinder.*

GENERAL STEAM NAVIGATION Co's S. S. "BITTERN"

JAN. 19TH 1899.
STEAM 150 LBS.
REVS. 80.

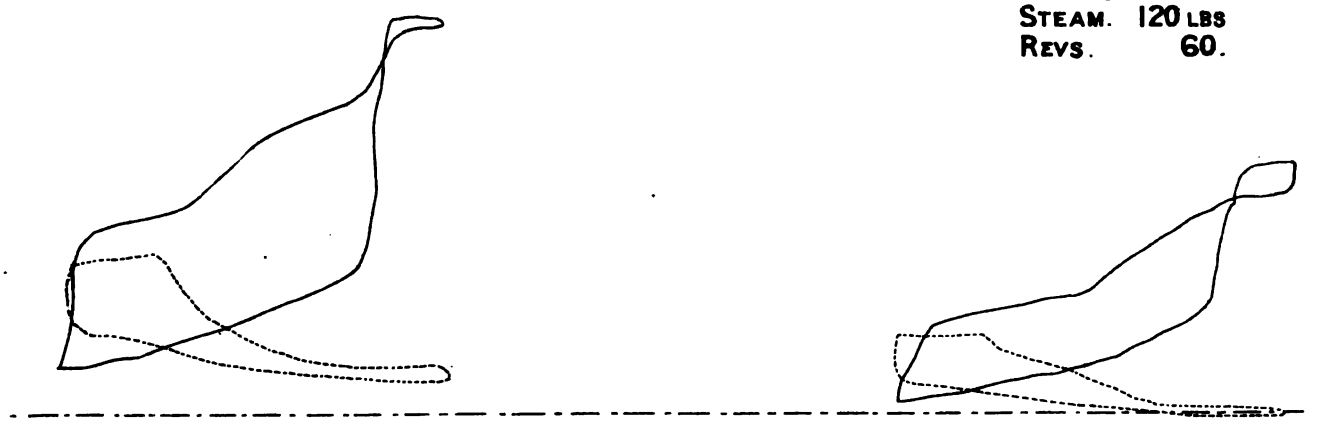
ENGINES FITTED BY MESSRS LAIRD BROTHERS.

AUG. 30TH 1900



MESSRS BAILEY & LEETHAM'S S. S. "SULTANA"

JAN. 16TH 1901.
STEAM. 120 LBS
REVS. 60.



DIAGRAMS TAKEN FROM
NEW TYPE, STEAM ASSISTANT CYLINDER.

FIG. 4.

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*Illustrate Professor W. E. Dalby's Paper on the balancing of reciprocating Parts of Engines, including the Effect of the connecting Rod.*

INITIAL CONDITIONS

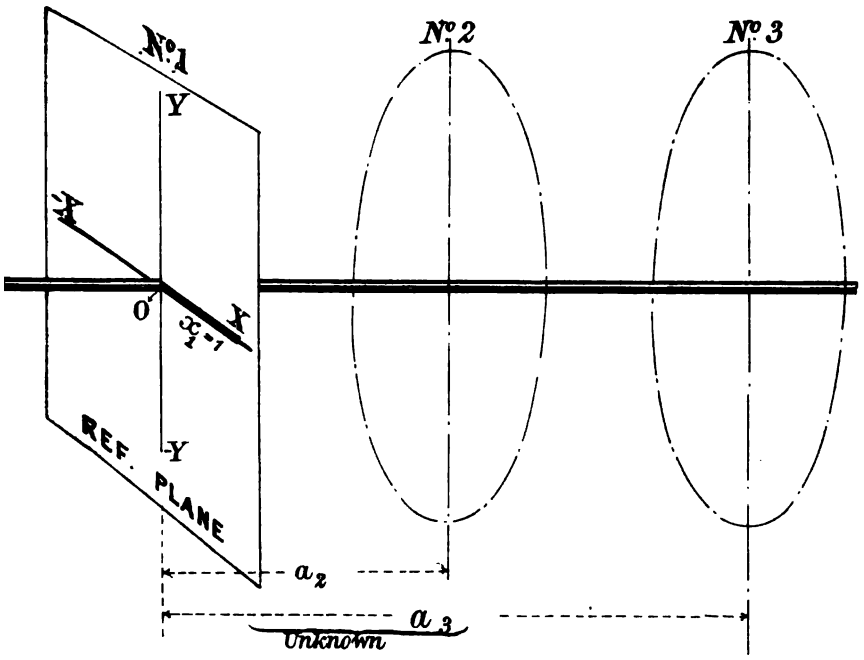


FIG. 5.

SOLUTION

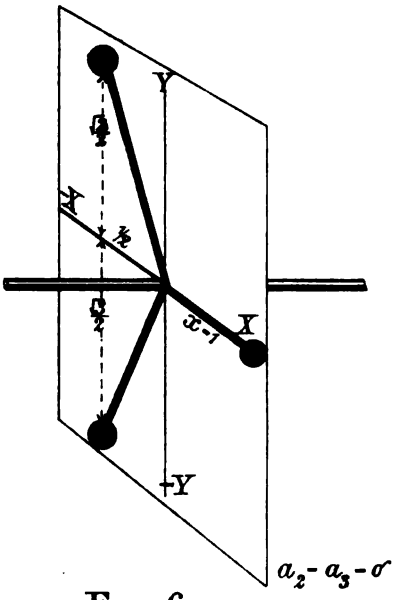


FIG. 6.

SOLUTION

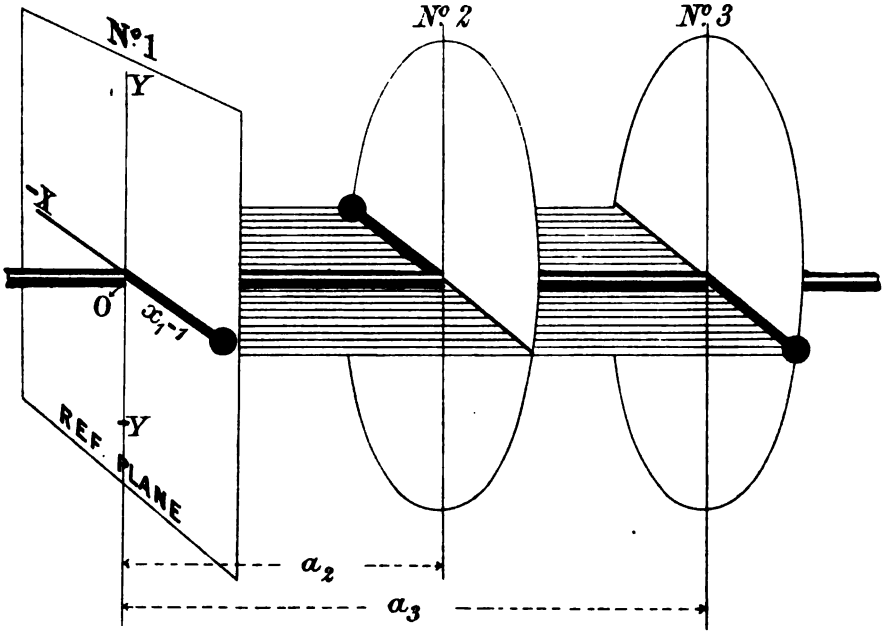


FIG. 7.

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*ustrate Professor W. E. Dalby's Paper on the balancing of reciprocating Parts of Engines, including the Effect of the connecting Rod.*

INITIAL CONDITIONS

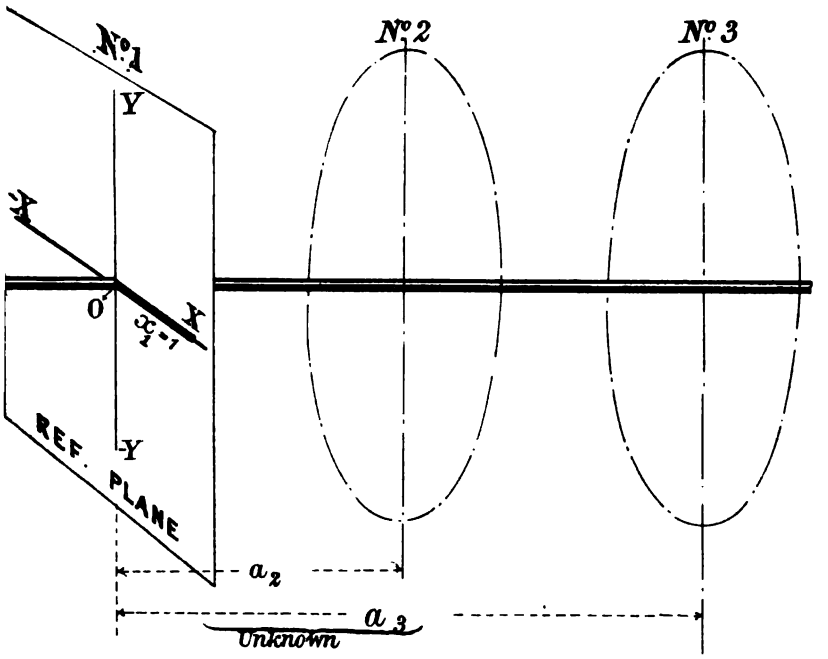


FIG. 5.

SOLUTION

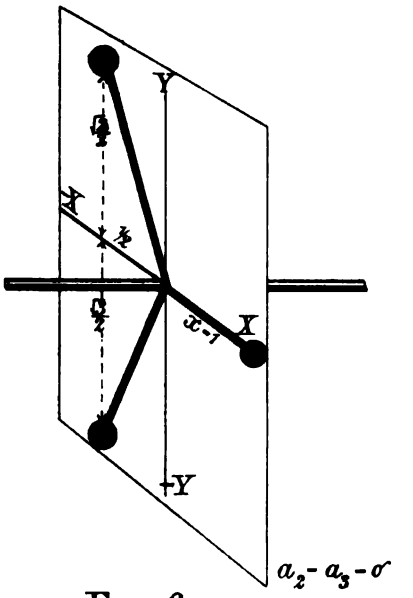


FIG. 6.

SOLUTION

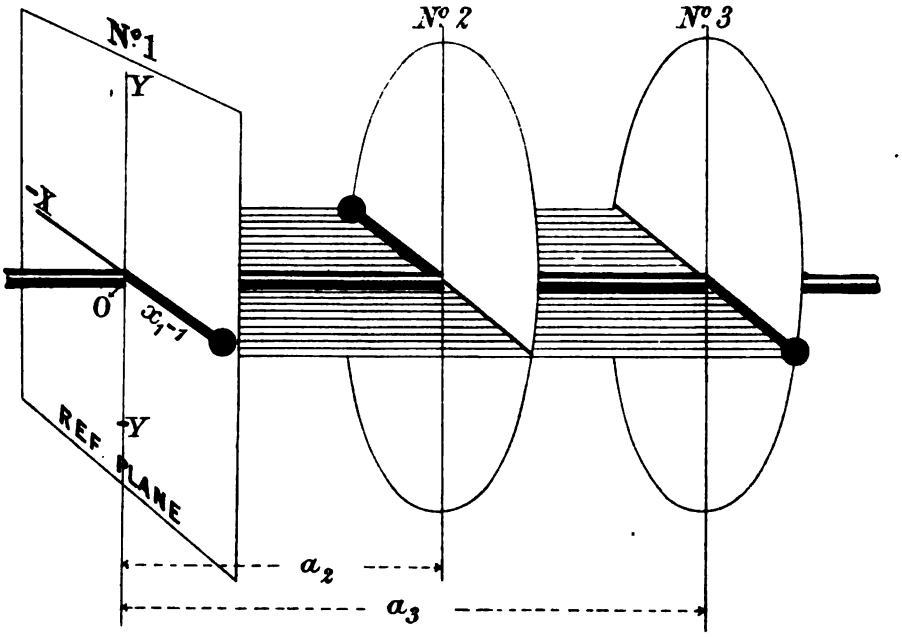


FIG. 7.

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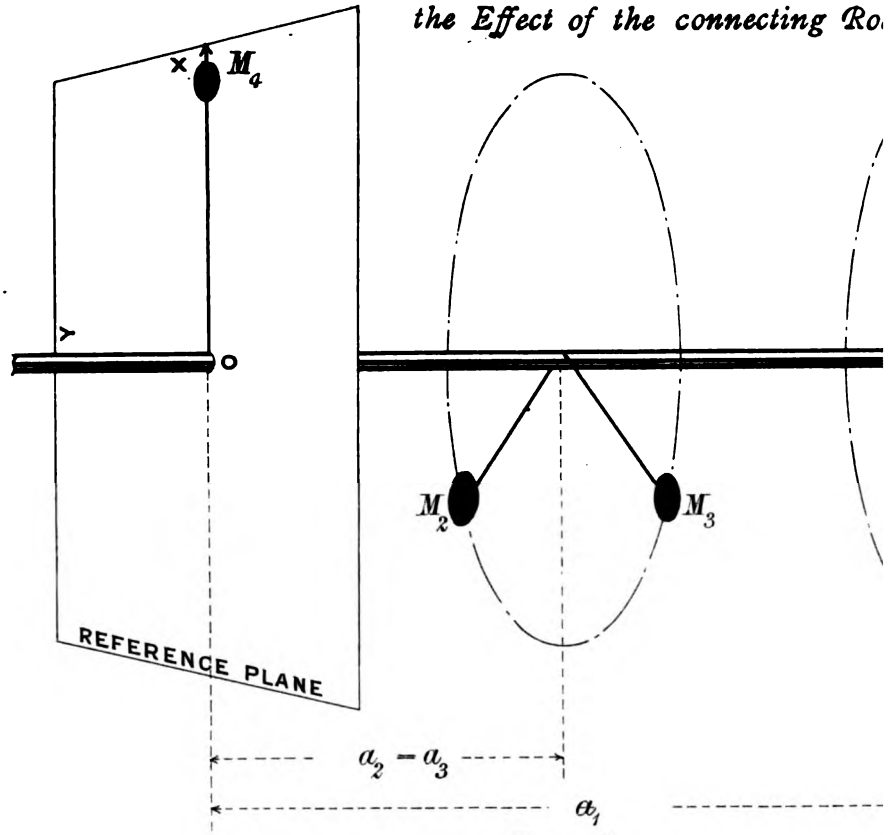


FIG. 8.

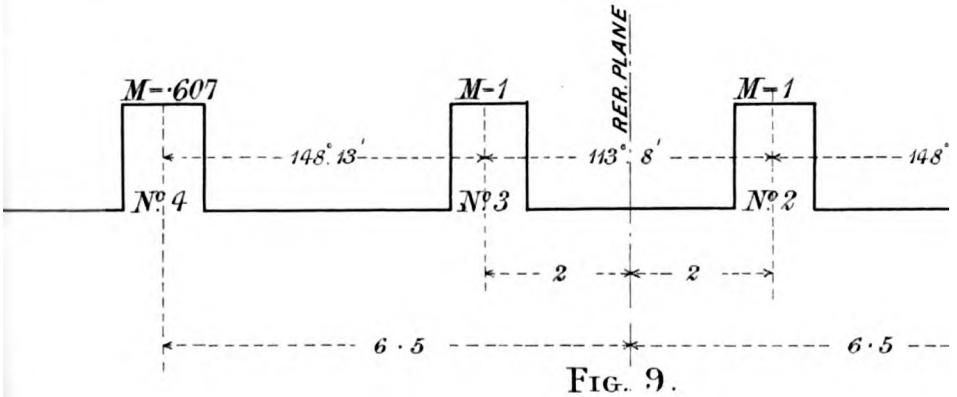


FIG. 9.

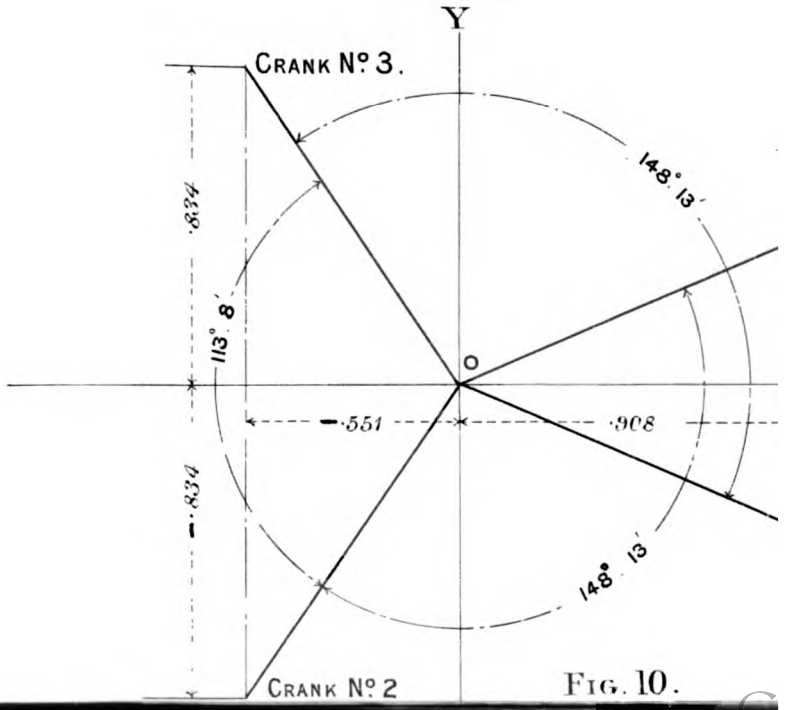


FIG. 10.

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*Illustrate Professor W. E. Dalby's Paper on the balancing of reciprocating Parts of Engines, including the Effect of the connecting Rod.*

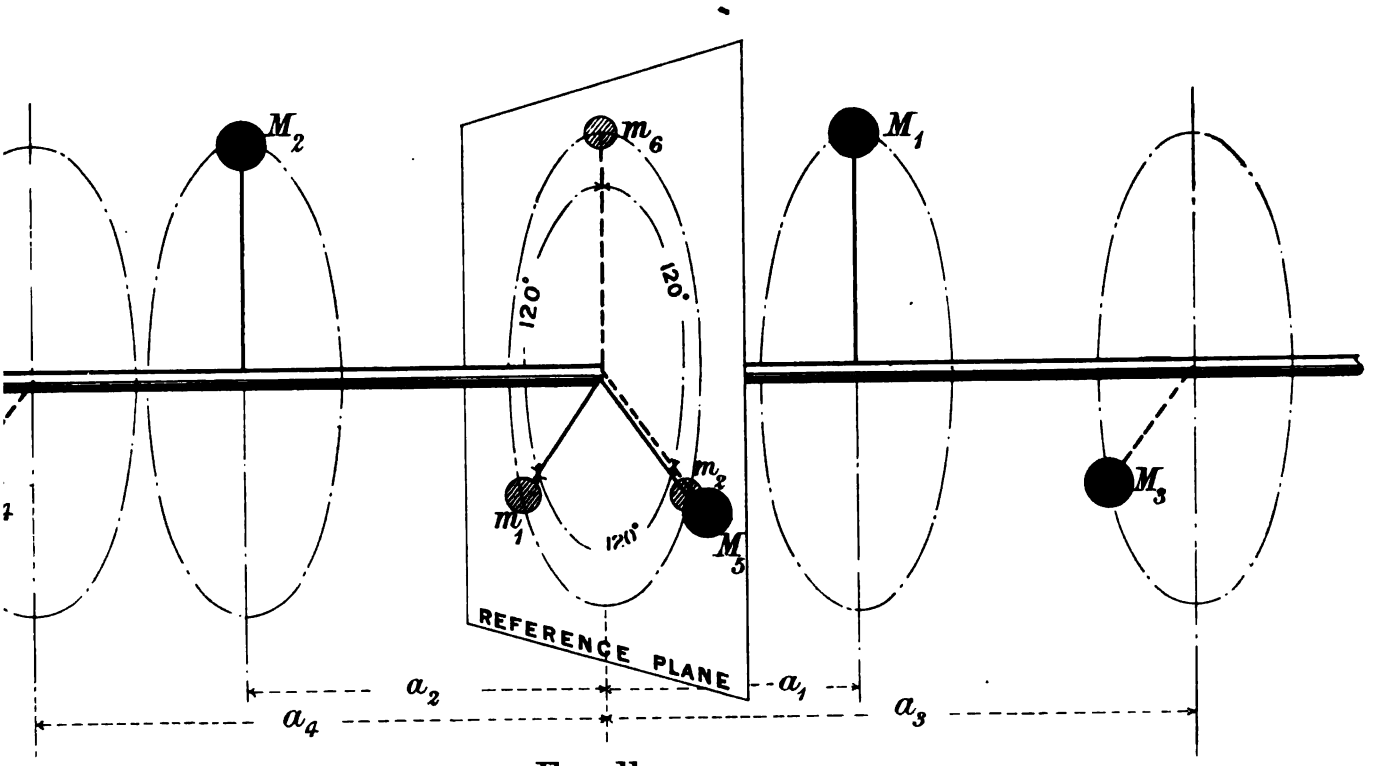


FIG. 11.

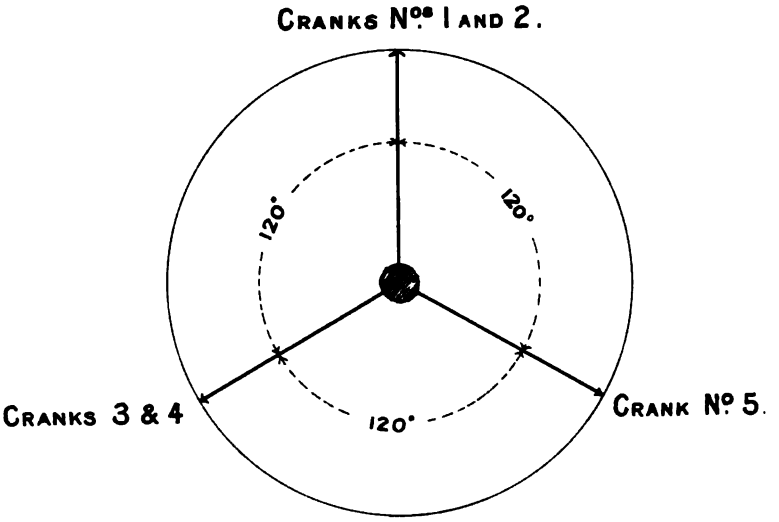


FIG. 12.

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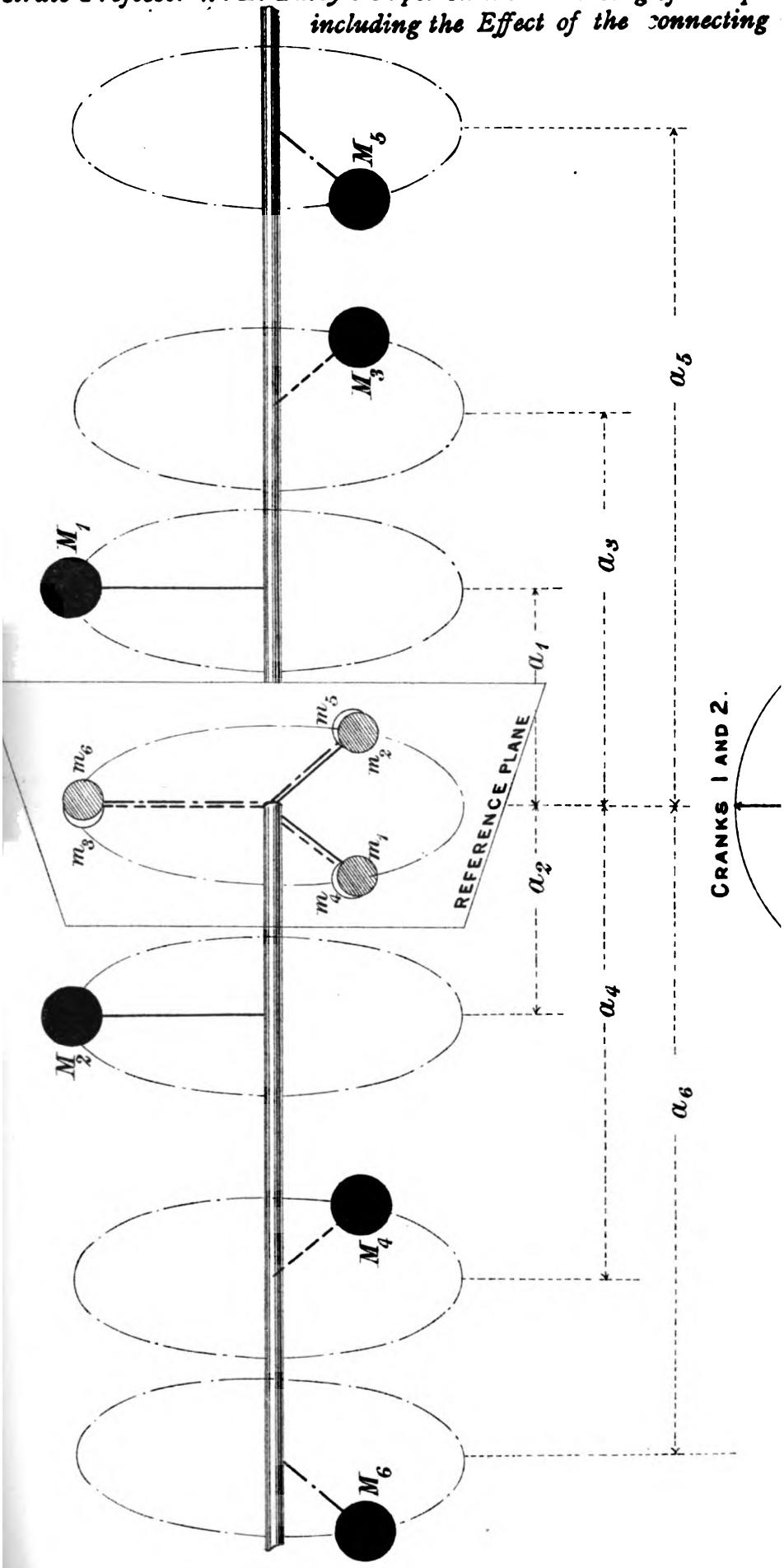
*strate Professor W. E. Dalby's Paper on the Balancing of recipro*
*including the Effect of the connecting*

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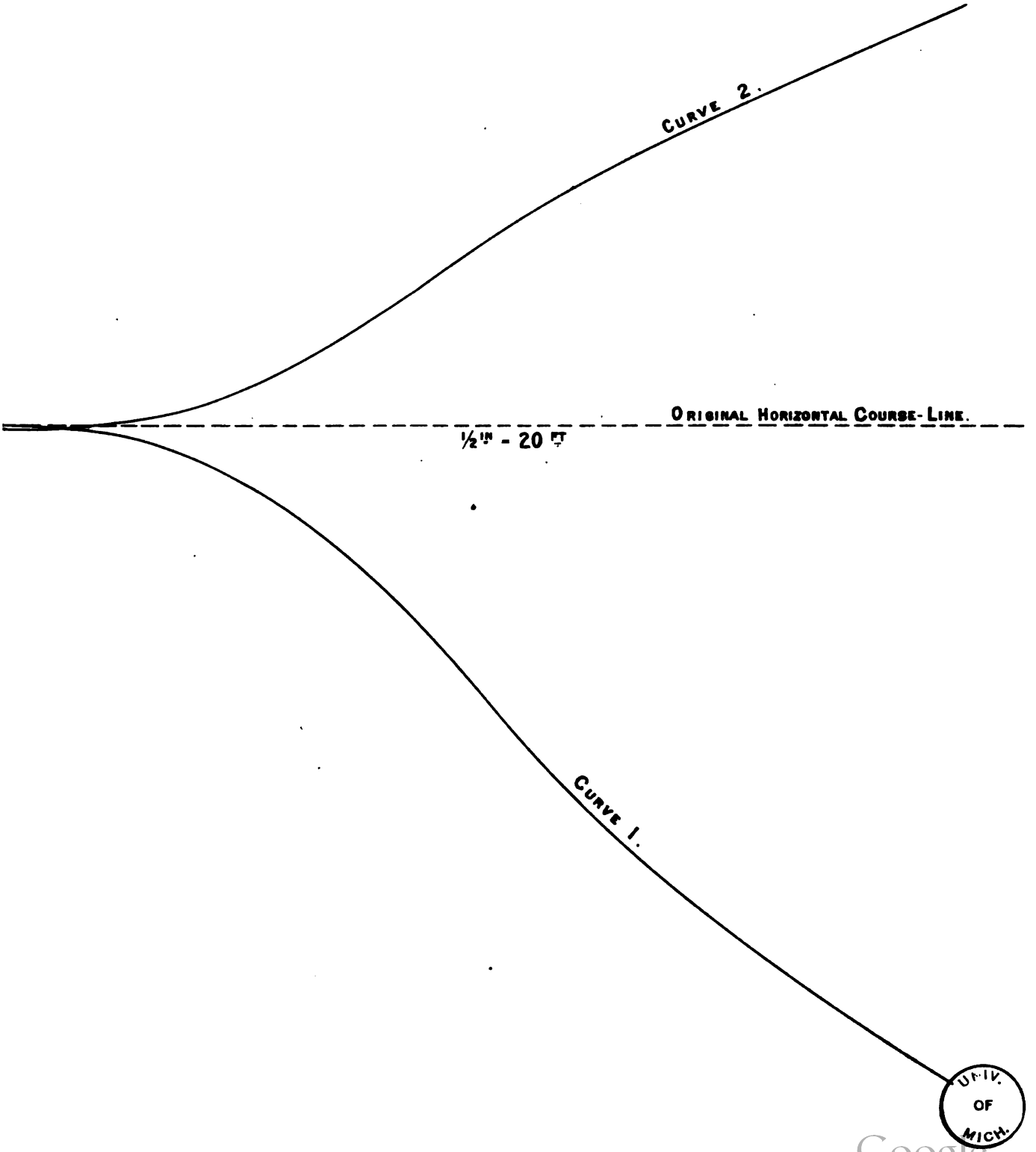
*To Illustrate Kaptain G. W. Hovgaard's Paper: The Motion of submarine Boats in the Vertical Plane.*

CURVES SHOWING PATH OF CENTRE OF GRAVITY OF A SUBMARINE BOAT, WHICH, BEING IN STEADY MOTION ALONG A HORIZONTAL LINE, IS DISTURBED BY:

1. A WEIGHT ADDED FORWARD
2. A WEIGHT ADDED AFT:

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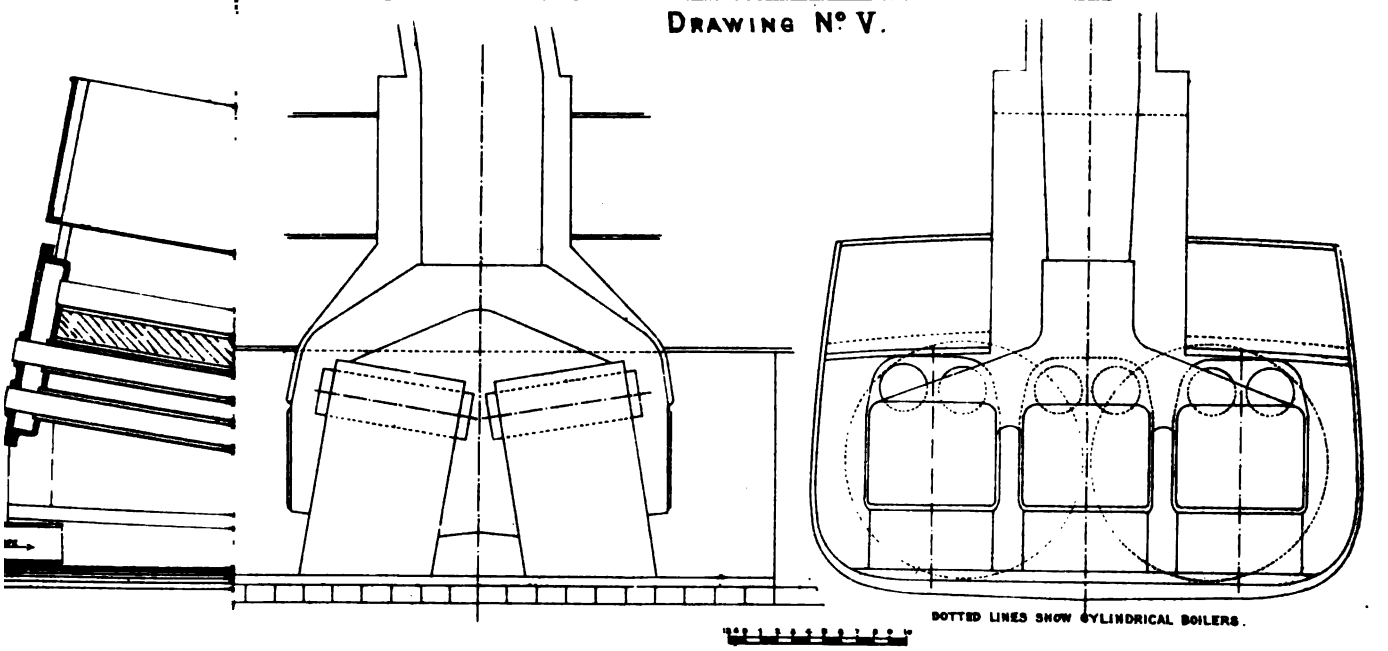
# Double Tube Boiler

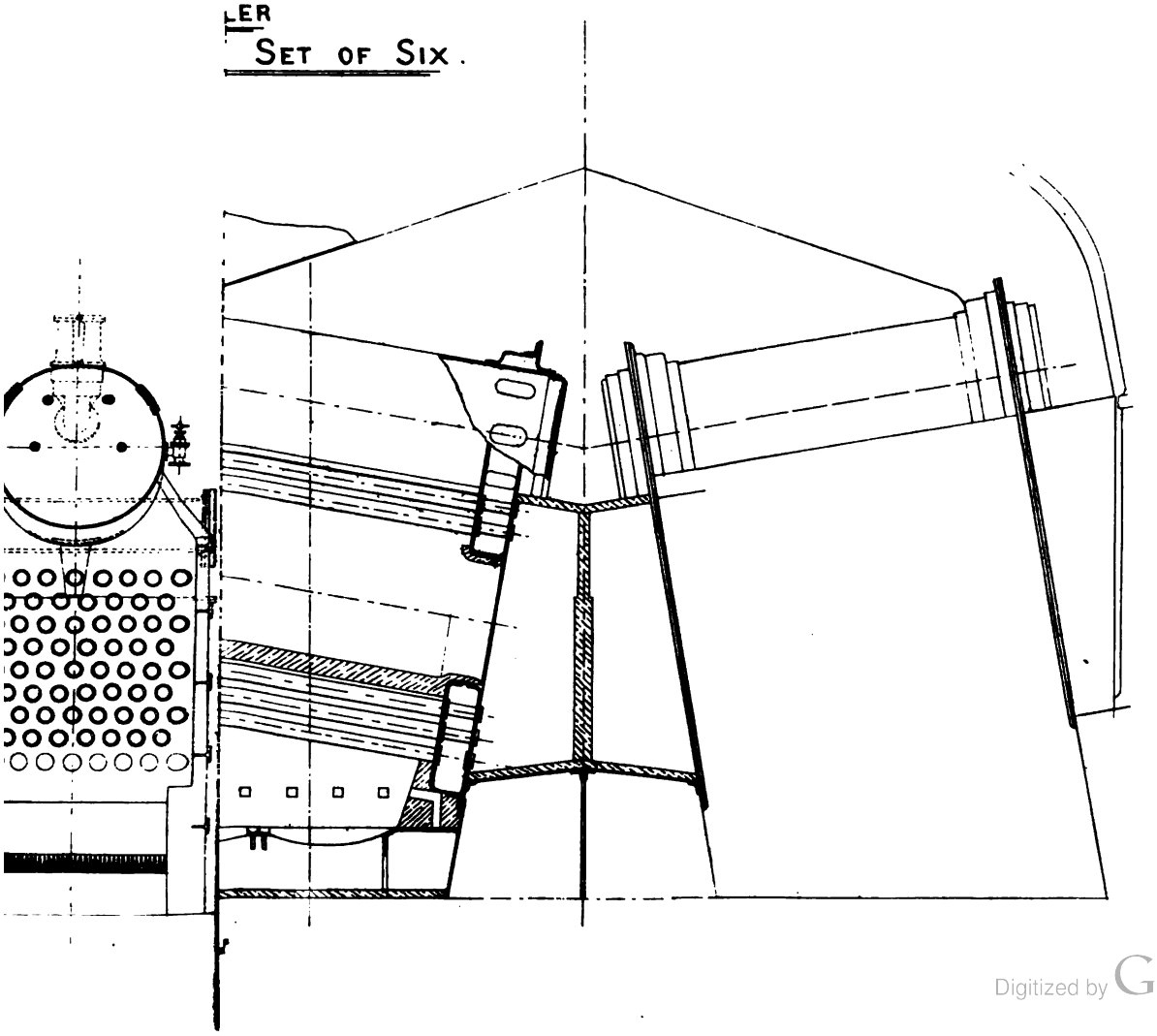
## Arrangement of Group of Six to Illustrate Table II.

### Drawing No. V.




LER

Set of Six.



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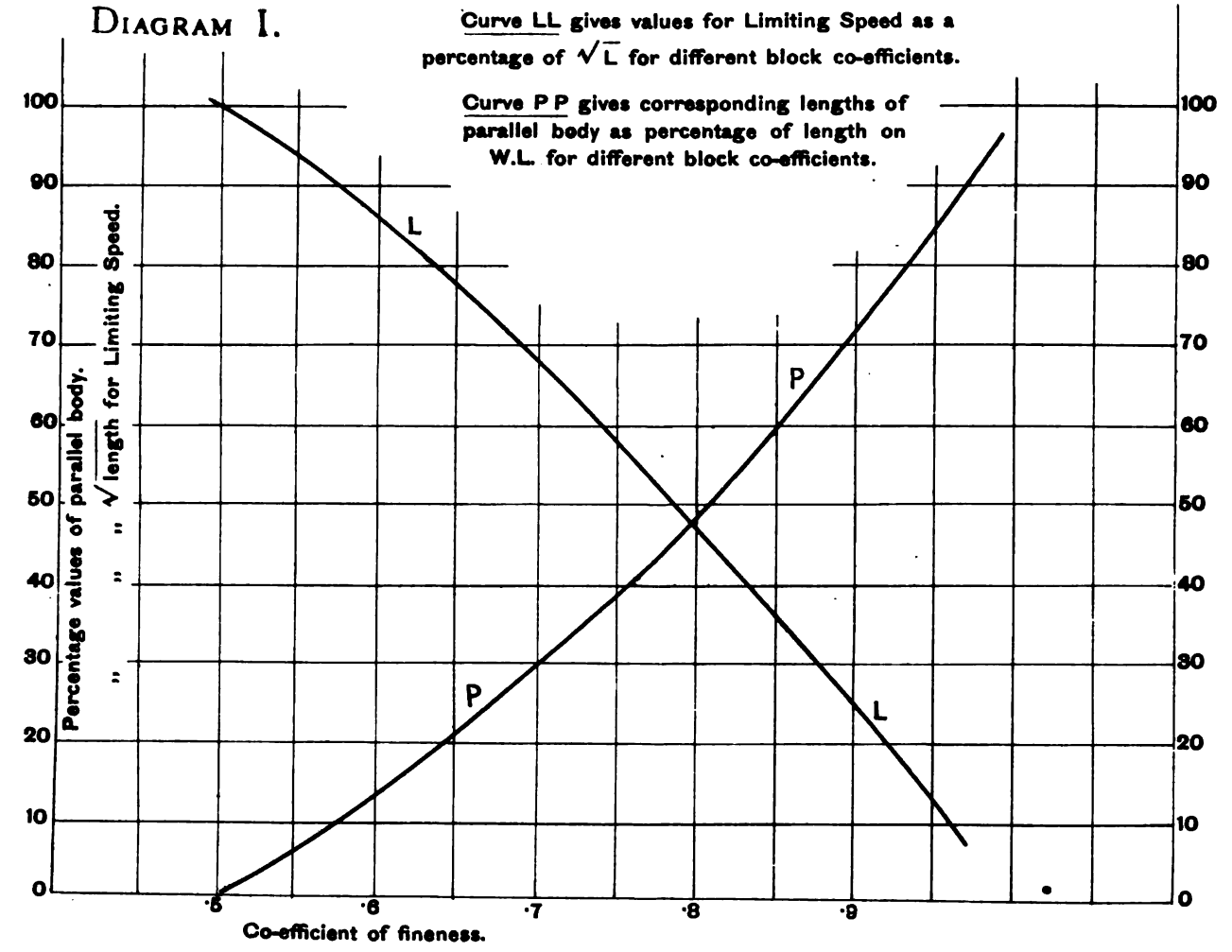


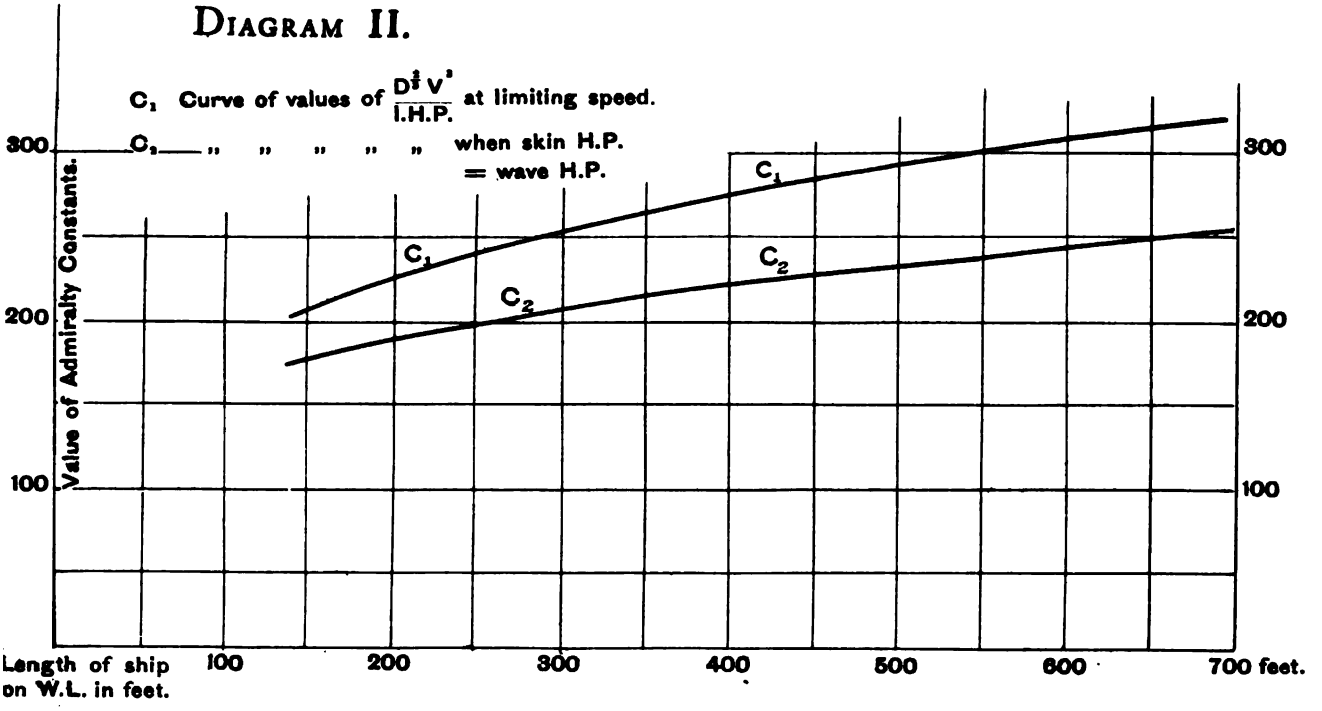


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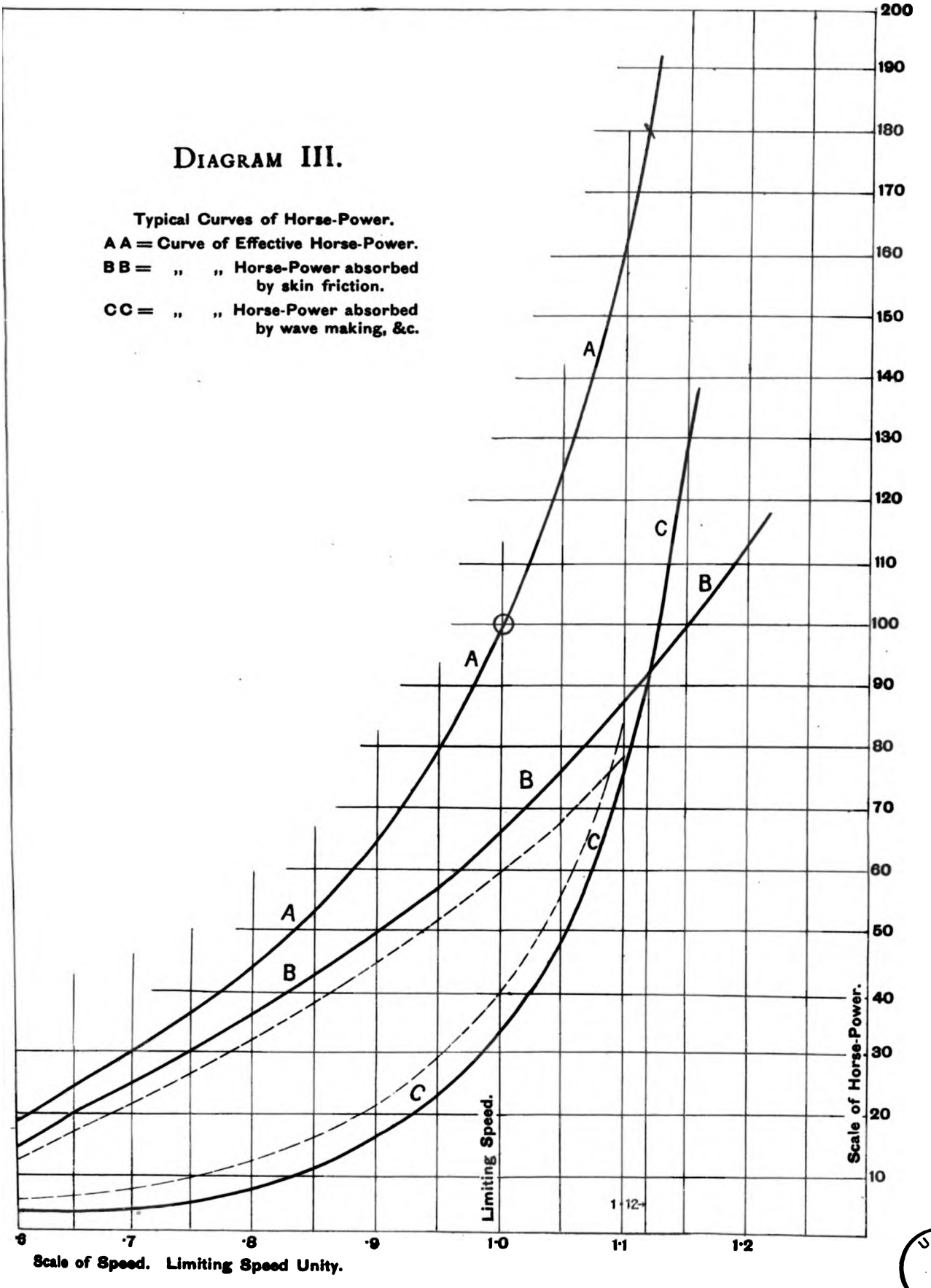
*rate Mr. E. H. Tennyson-d'Eyncourt's Paper on the Limits of Economical Speed of Ships.*

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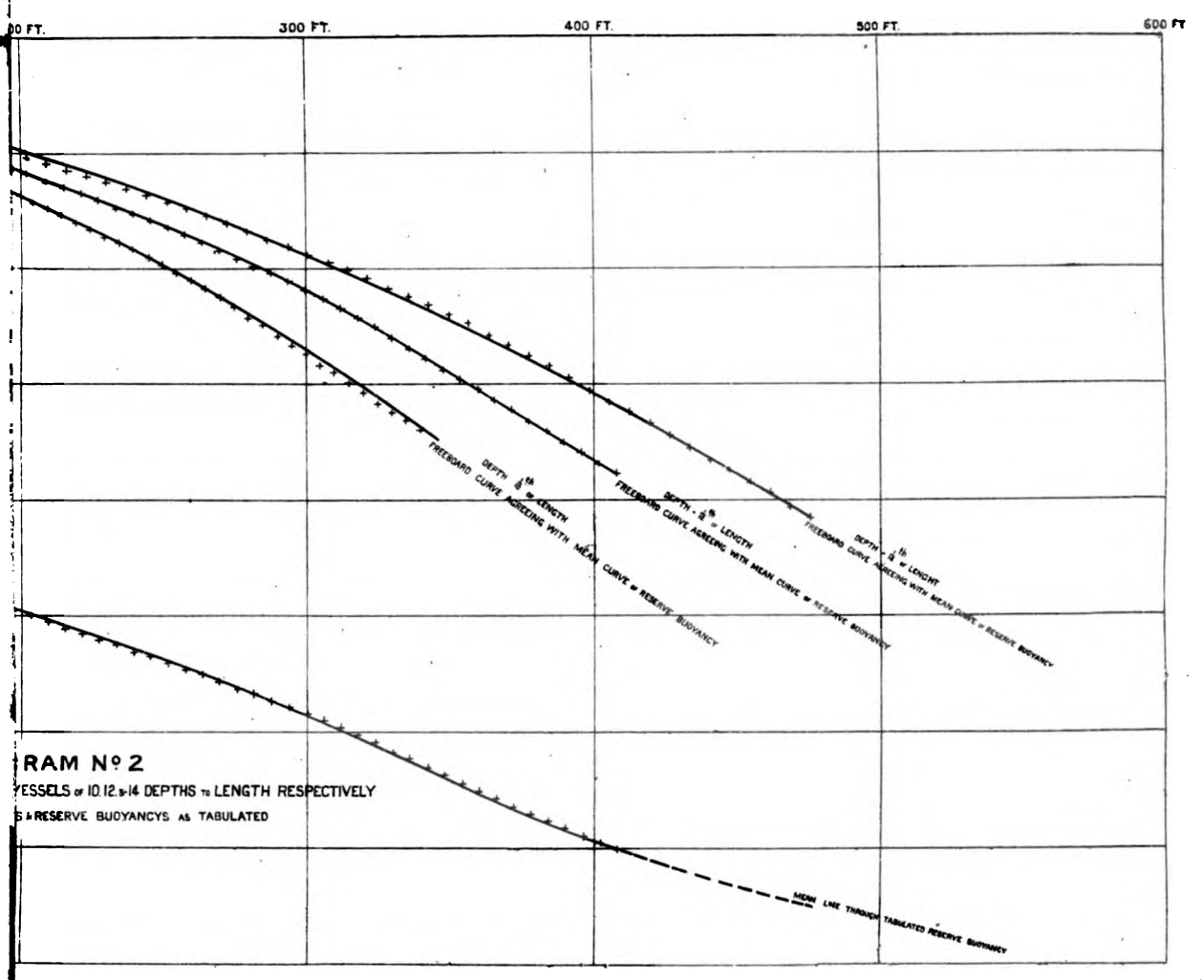
Trans.




RAM Nº 2
ESSELS OF 10.12 & 14 DEPTHS TO LENGTH RESPECTIVELY
S & RESERVE BUOYANCYS AS TABULATED

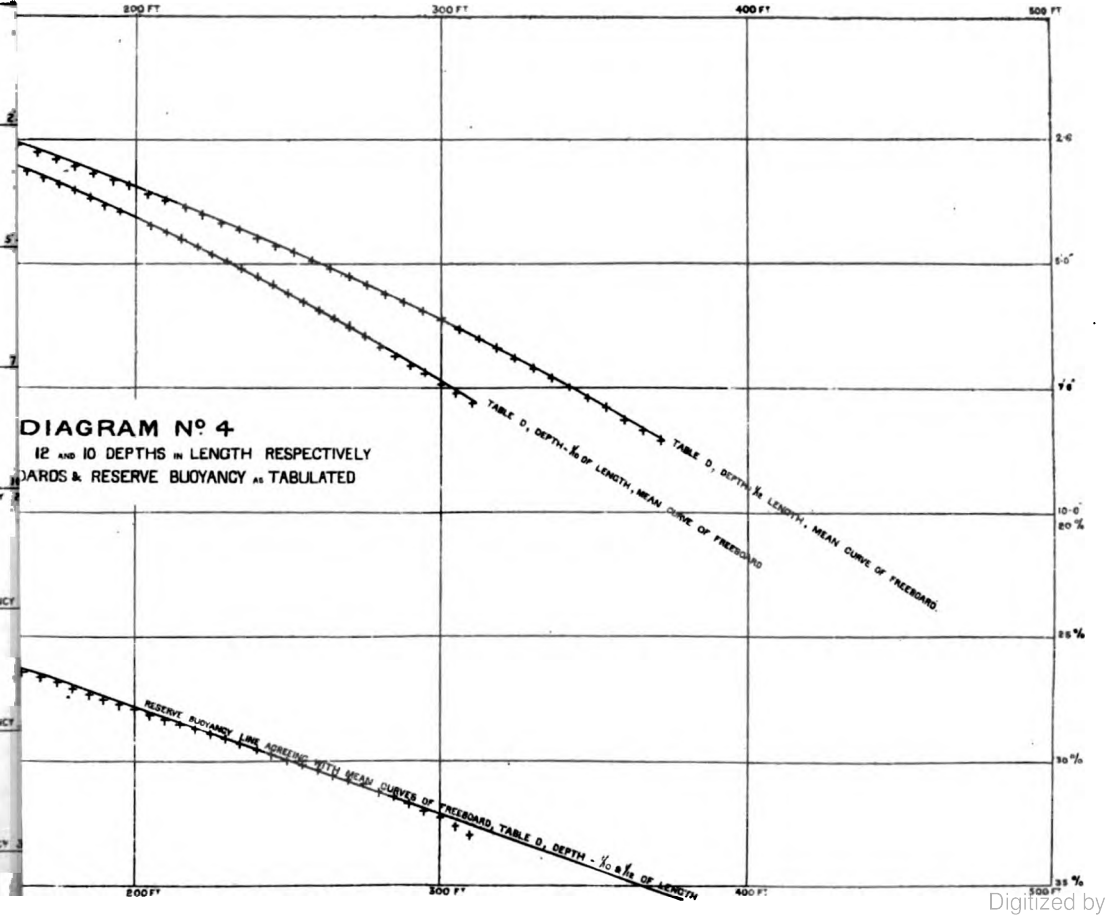



DIAGRAM Nº 4
12 AND 10 DEPTHS IN LENGTH RESPECTIVELY
ARDS & RESERVE BUOYANCY AS TABULATED

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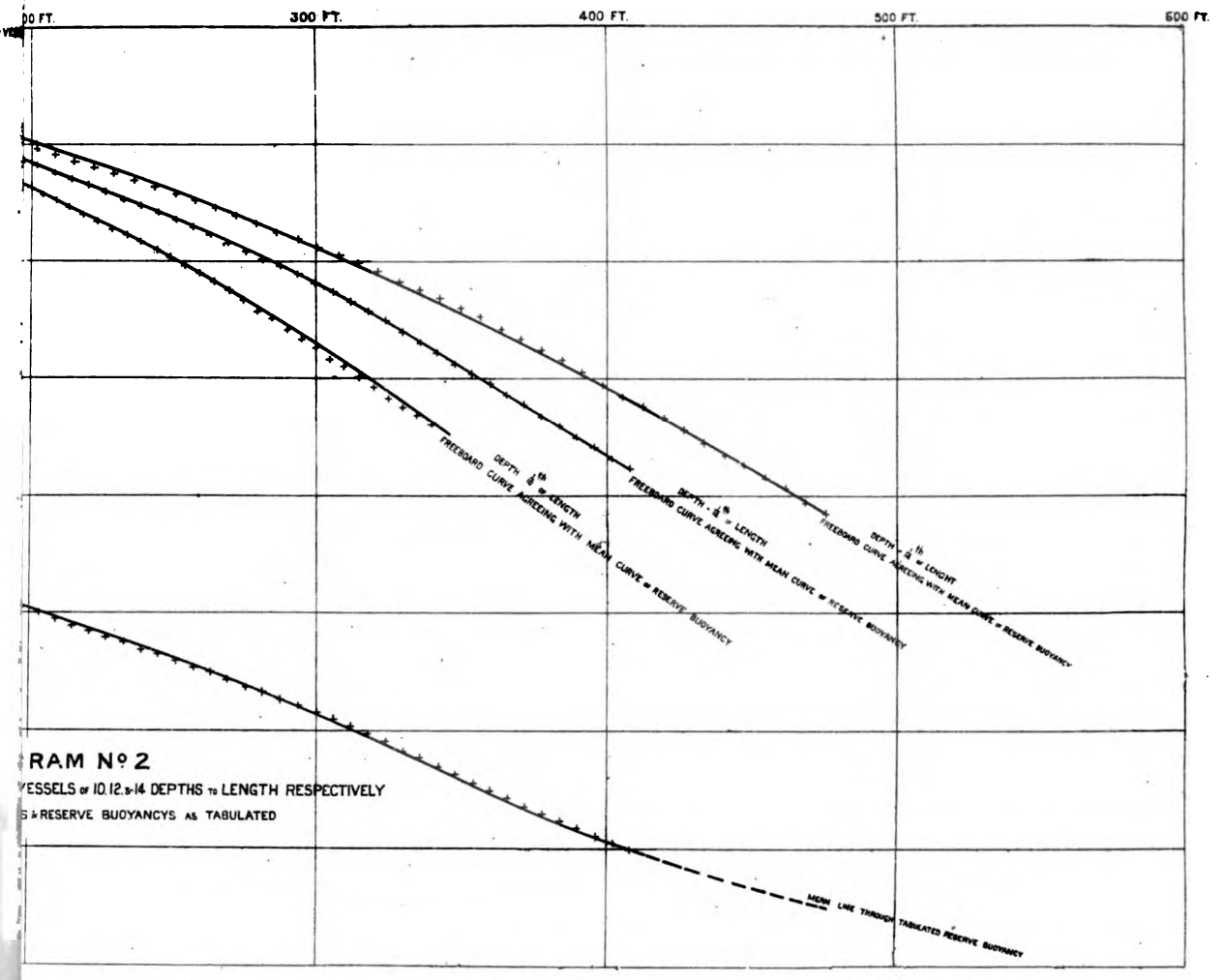


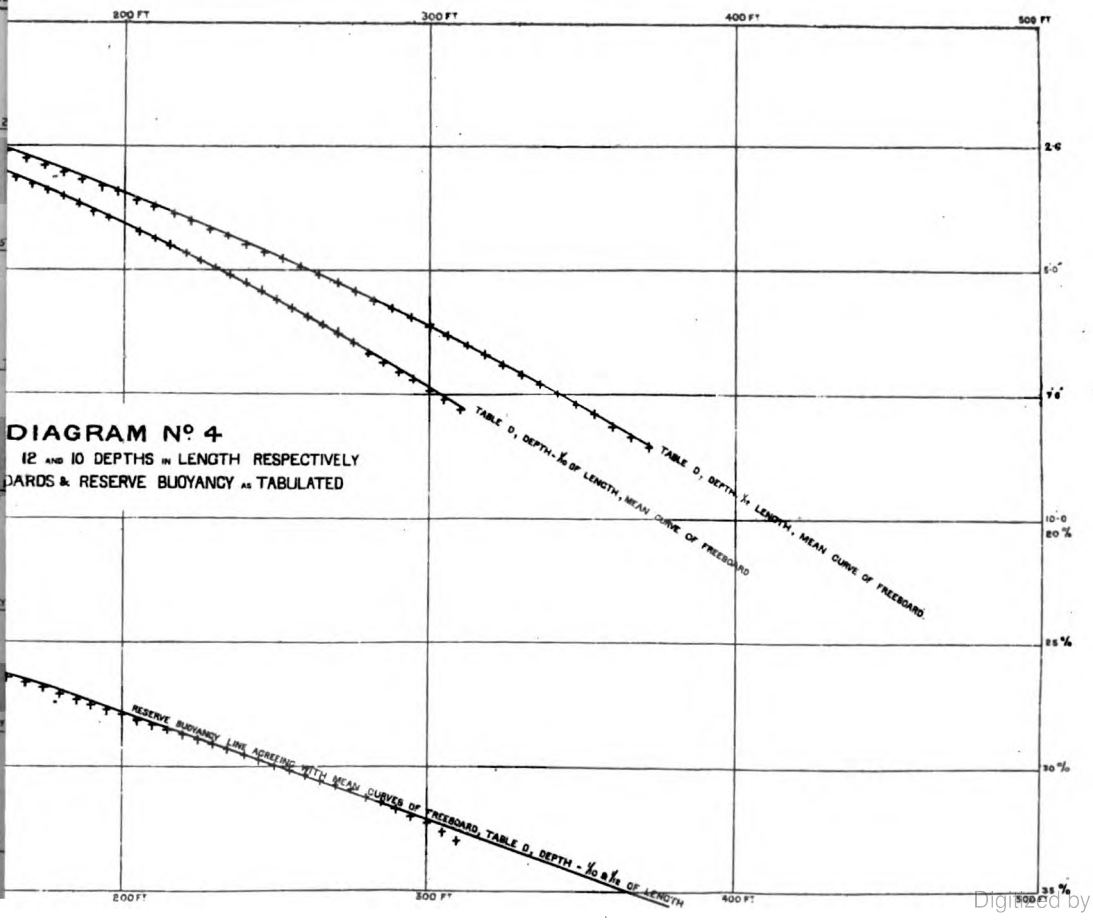


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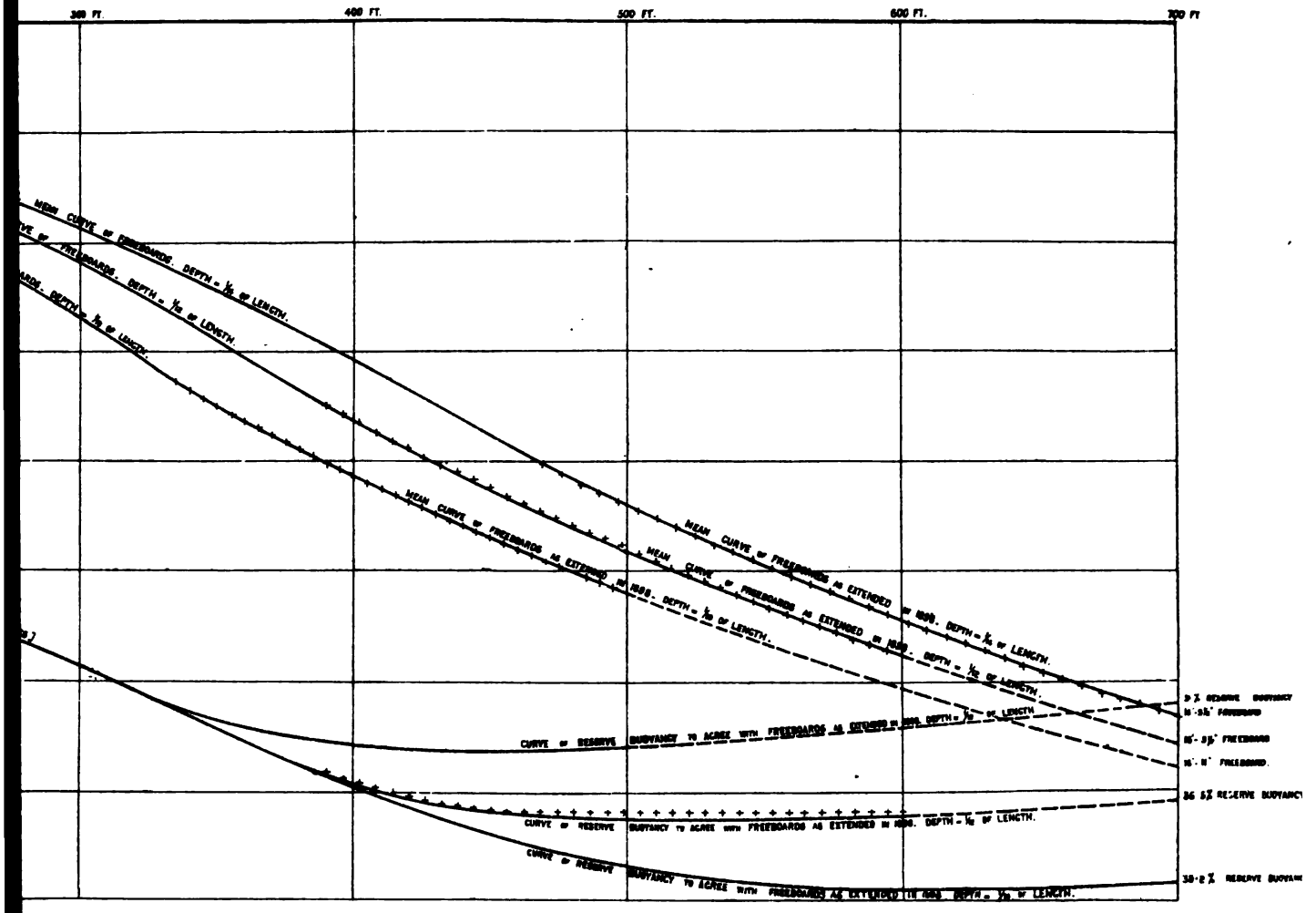


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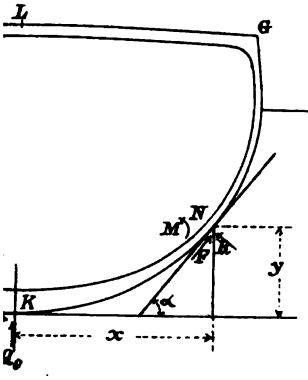
FIG. 1.



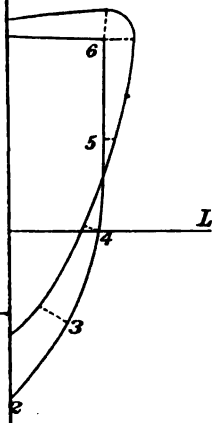




FIG. 5.

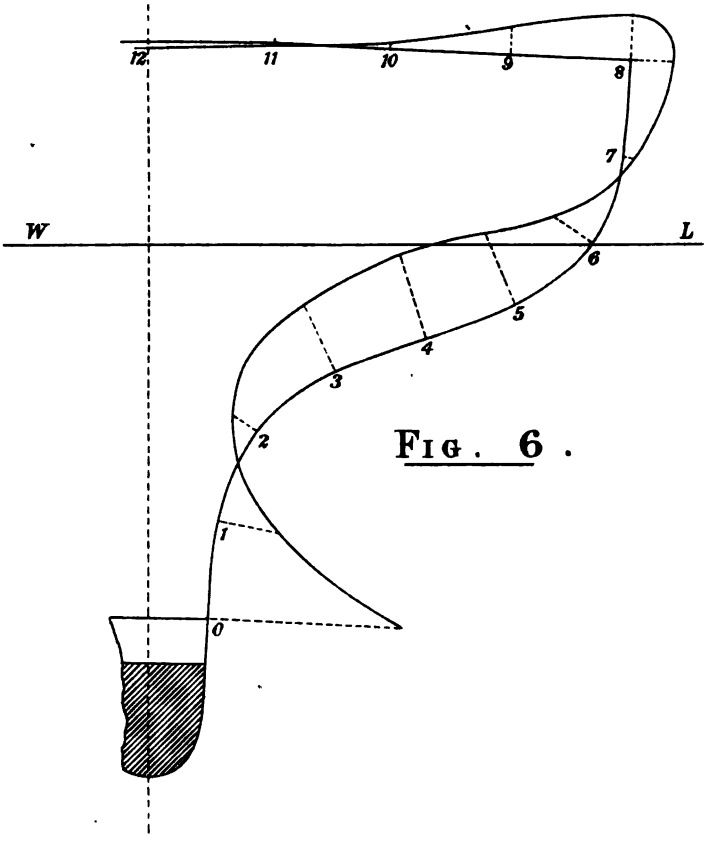



FIG. 6.

FIG. 8.

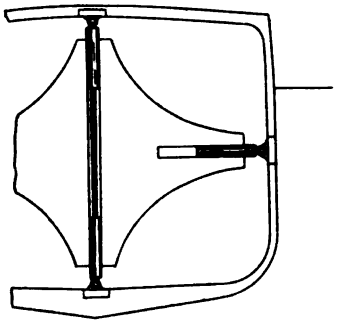


FIG.

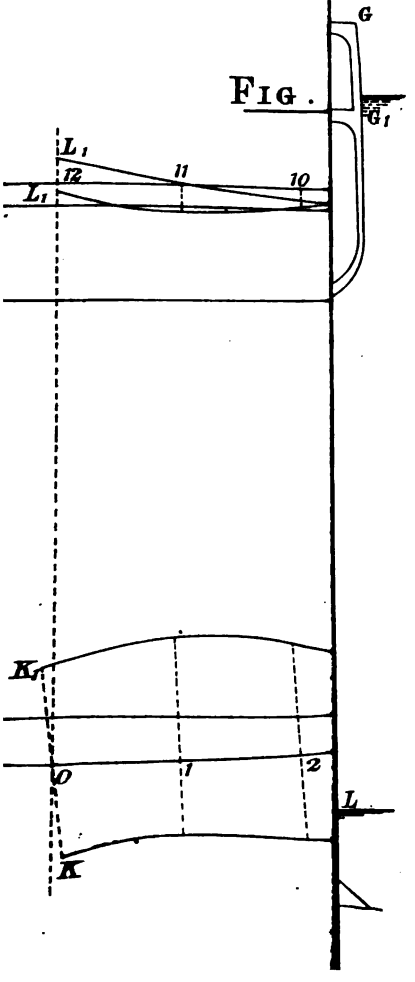
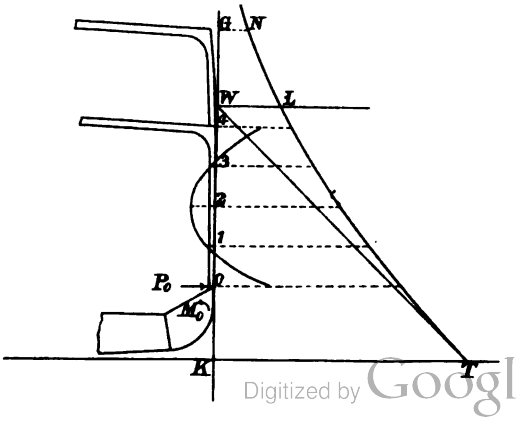



FIG. 10.

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the Vibration Problem.

# ENGINE.

L.P. PISTON
FIG. 3.

1ST I.P. PISTON.
FIG. 2.

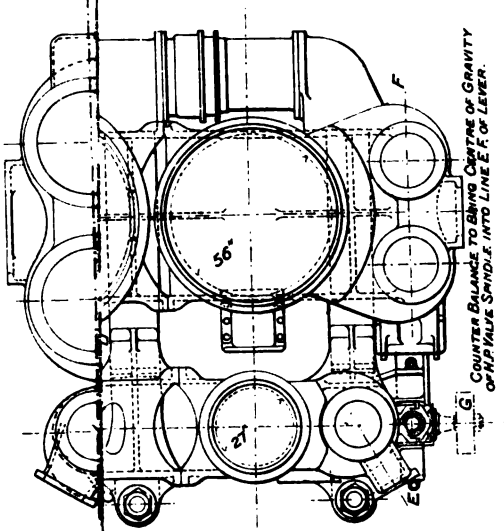



FIG. 5.

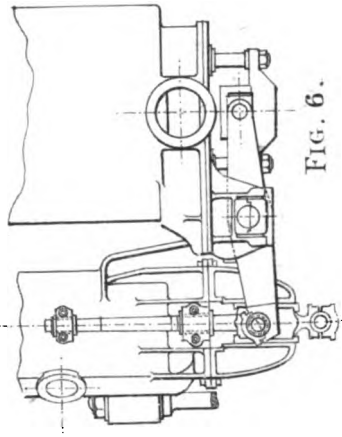


FIG. 6.

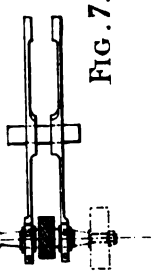


FIG. 7.

ALTERNATIVE PLANS.

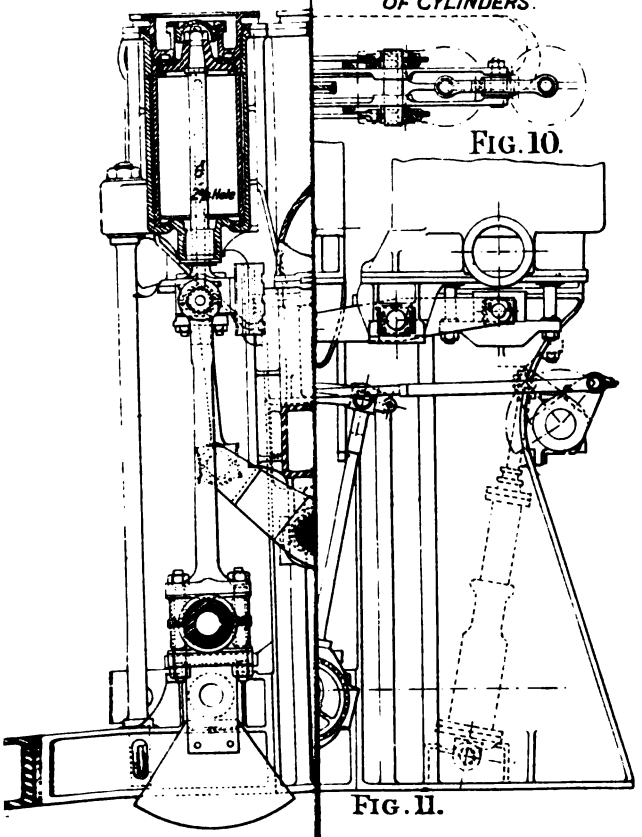
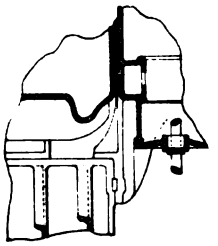
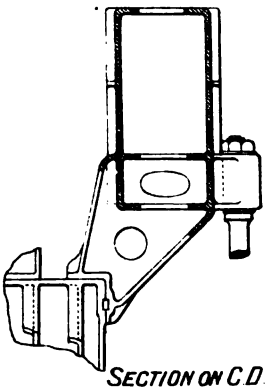



FIG. 10.

FIG. 11.



SECTION ON A.B.
FIG. 12.



SECTION ON C.D.
FIG. 13.

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# BALANCED MARINE ENGINE.

CYLINDERS ..... 27" 39" 56" 80"
STROKE ..... 48"
BOILER PRESSURE ..... 200 LBS.

ARRANGEMENT OF STEAM CONNECTIONS.

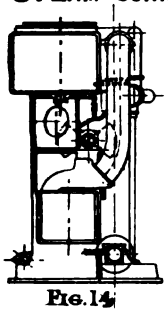

FIG. 14.

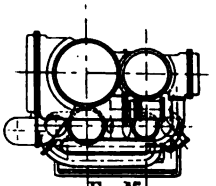

FIG. 15.

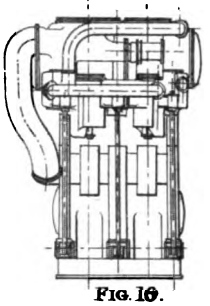

FIG. 16.

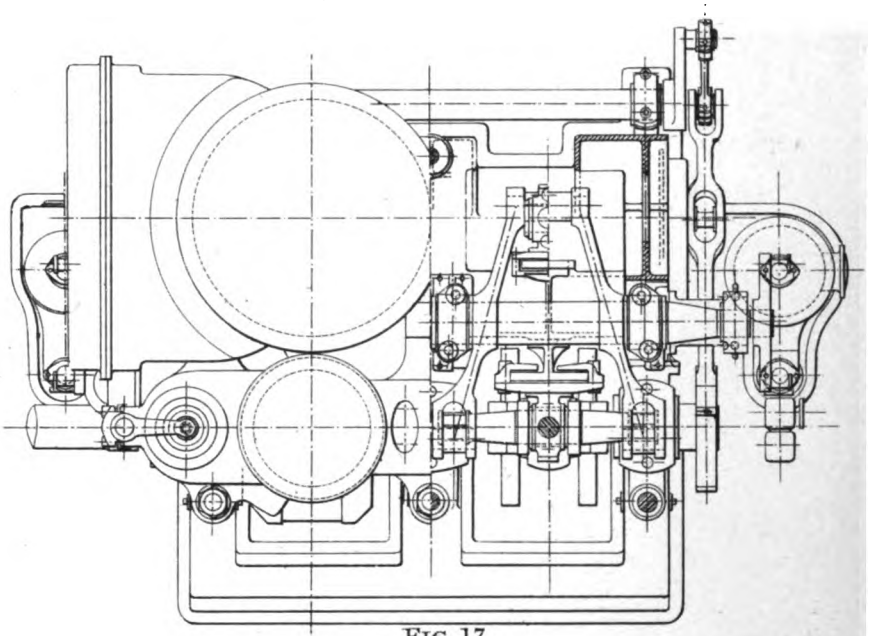

FIG. 17.

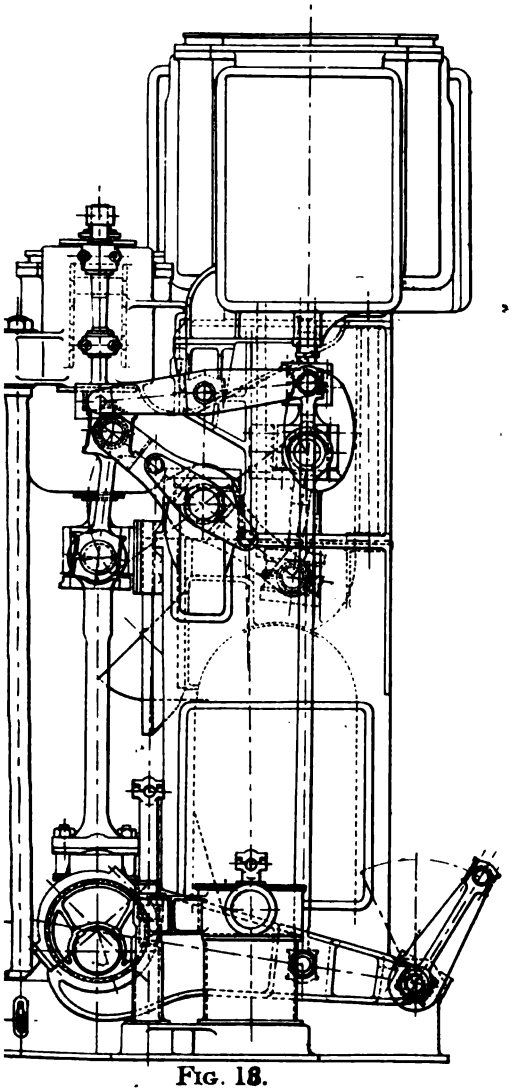

FIG. 18.

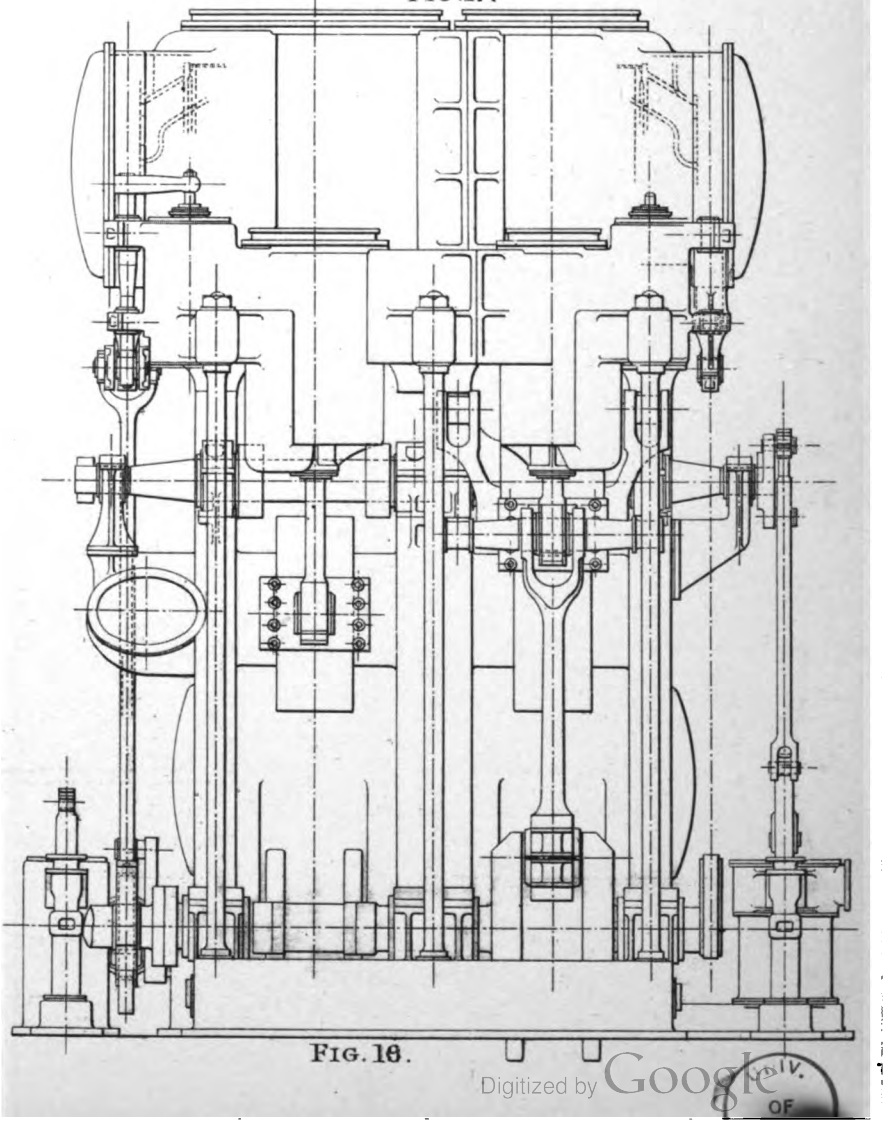

FIG. 19.

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*Illustrate Professor J. H. Biles' Paper: Standardising the Results of Ship Calculations.*

## TABLE. A.

### PARTICULARS.

CEMENT TABLE OF

| | FEET | INCHES | | FEET |
|---|---|---|---|---|
| LENGTH BETWEEN PERPENDICULARS. (L) | | | WATER LINES APART. | |
| BREATH MOULDED. (B) | | | ORDINATES APART. | |
| DEPTH Do. (D) | | | FOREMOST ORDINATE FROM FORE PERPENDICULARS. | |
| DEPTH OF KEEL | | | AFTER ORDINATE FROM AFT Do. | |

RISE OF FLOOR

| WATER LINES / No OF ORDS | KEEL | FT | FT | FT | FT | FUNCT OF AREAS | MULTIPLES OF AREAS | MOMENTS | FT | FT | FT | FUNCT OF AREAS | MULTIPLES OF AREAS | MOMENTS | FT | FT | FT | FUNCT OF AREAS | MULTIPLES OF AREAS | MOMENTS |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| SHIPS MULTS | ½ | 2 | 1½ | 4 | 1 | | | | 1 | 4 | 1 | | | | 1 | 4 | 1 | | | |
| | ½ | 2 | 1½ | 4 | 1 | | | | 1 | 4 | 1 | | | | 1 | 4 | 1 | | | |
| | 0 | ½ | 1 | 2 | 3 | | | + | 3 | 4 | 5 | | | | 9 | 10 | 11 | | | |
| | | | | | | * | | | | | | | | | | | | | | |

DISPLACEMENT MULTIPLIER $\frac{*}{3} \times \frac{\dagger}{3} \times \frac{2}{35}$

* WATER LINES APART.
† ORDINATES APART.

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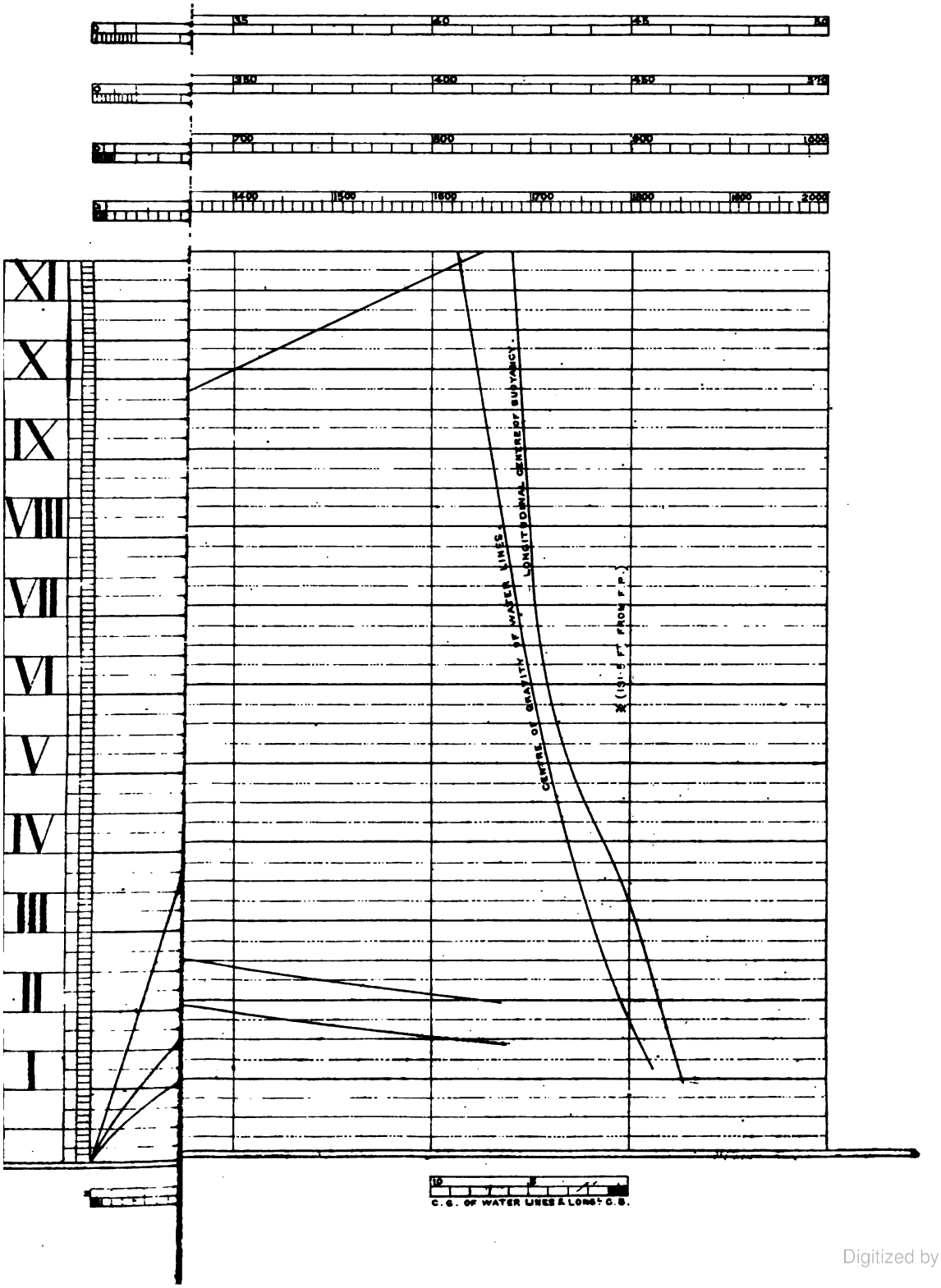
its of Ship Calculations.

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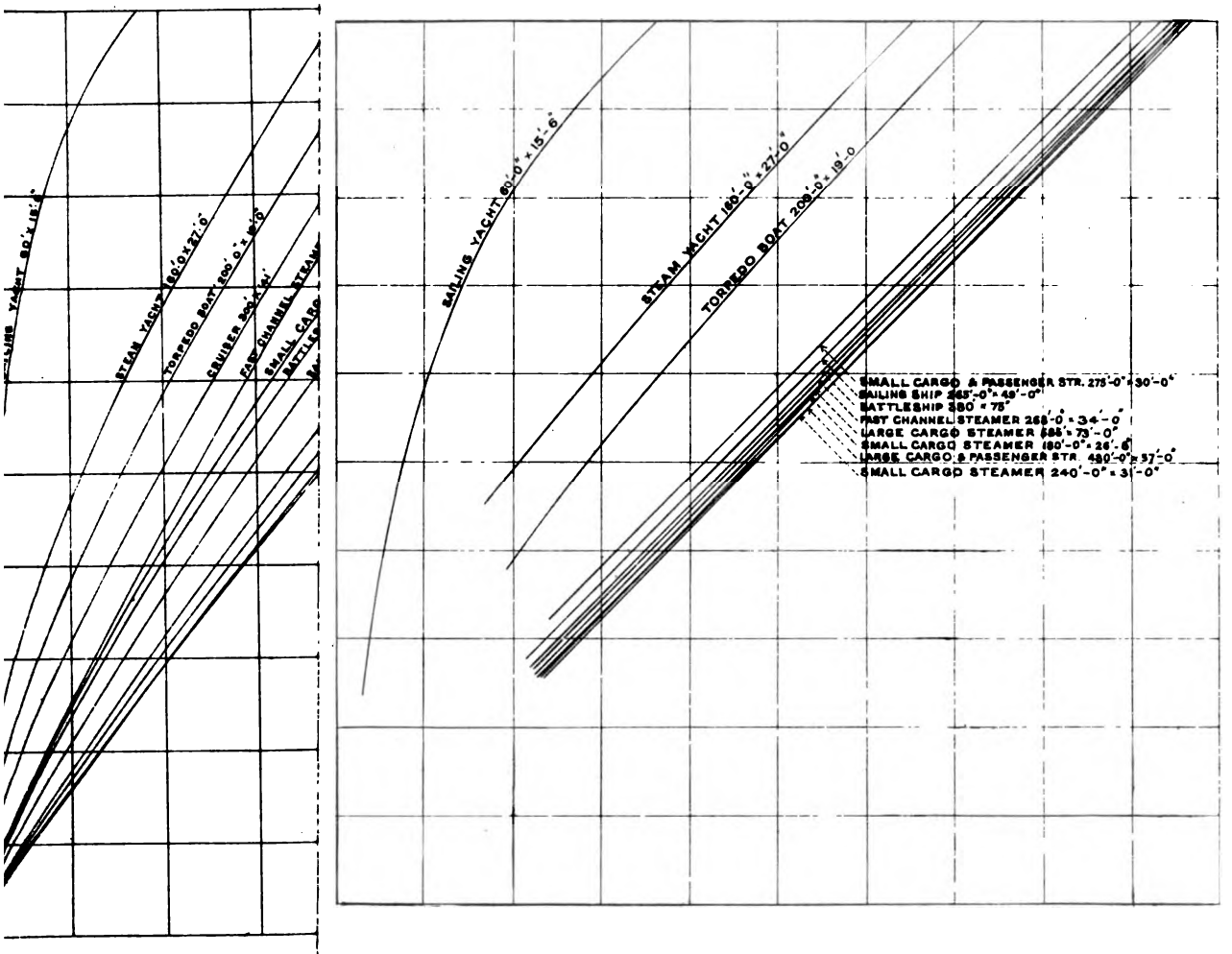
lations.

# Fig. 4

## Areas of Midship Sections.

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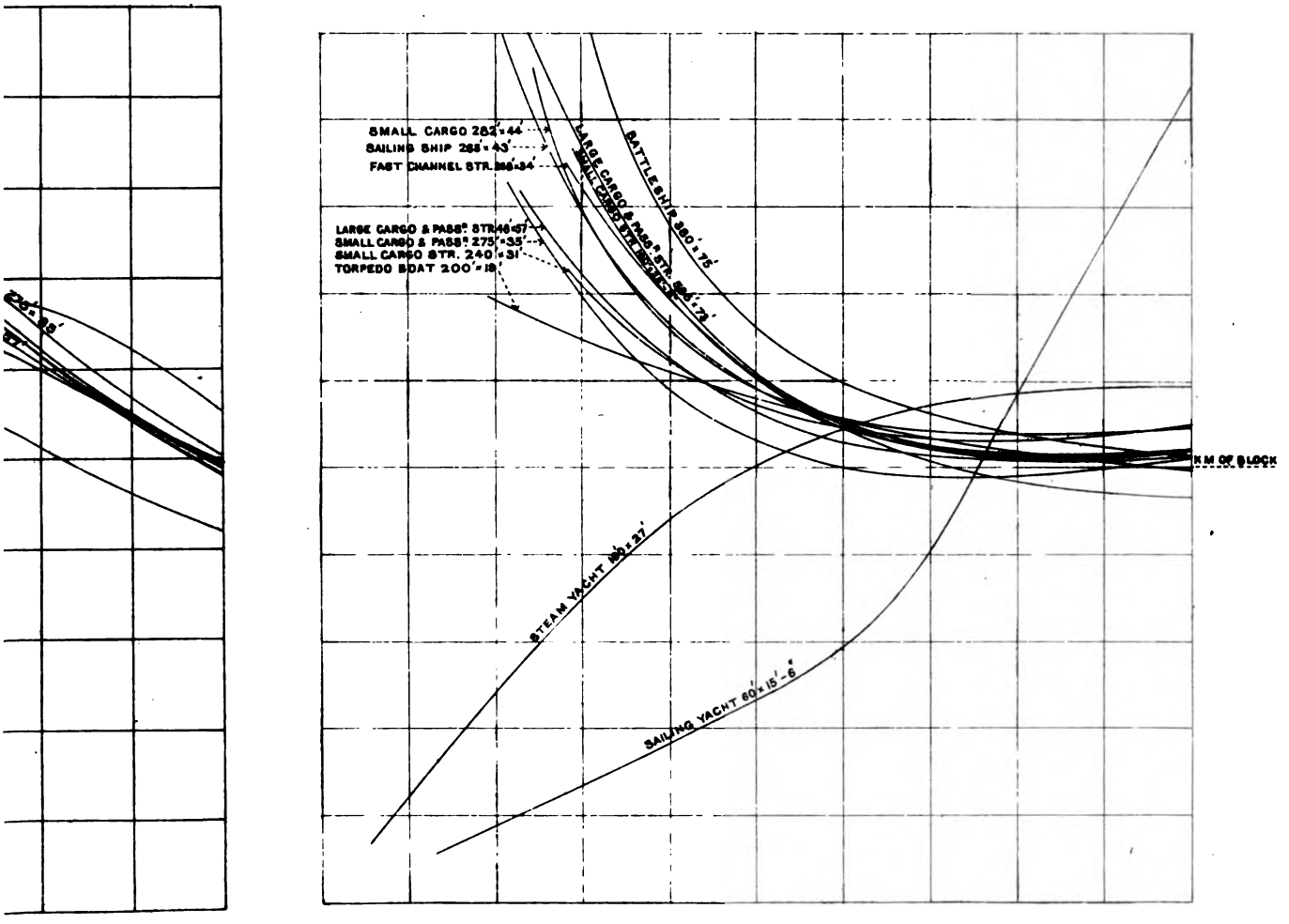
# Fig. 8.

## Transverse Metacentres KM's

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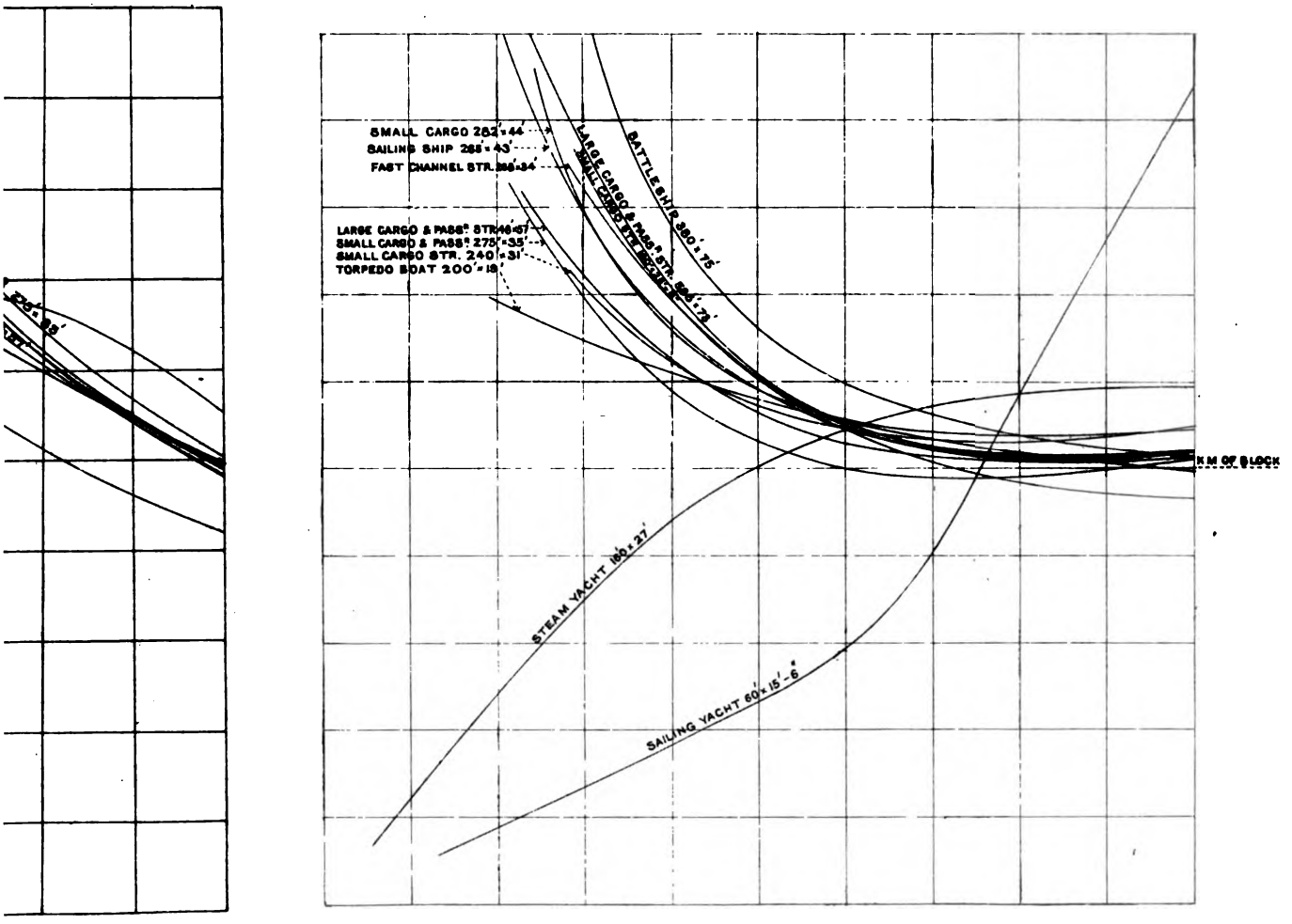
# Fig. 8.

## Transverse Metacentres KM's

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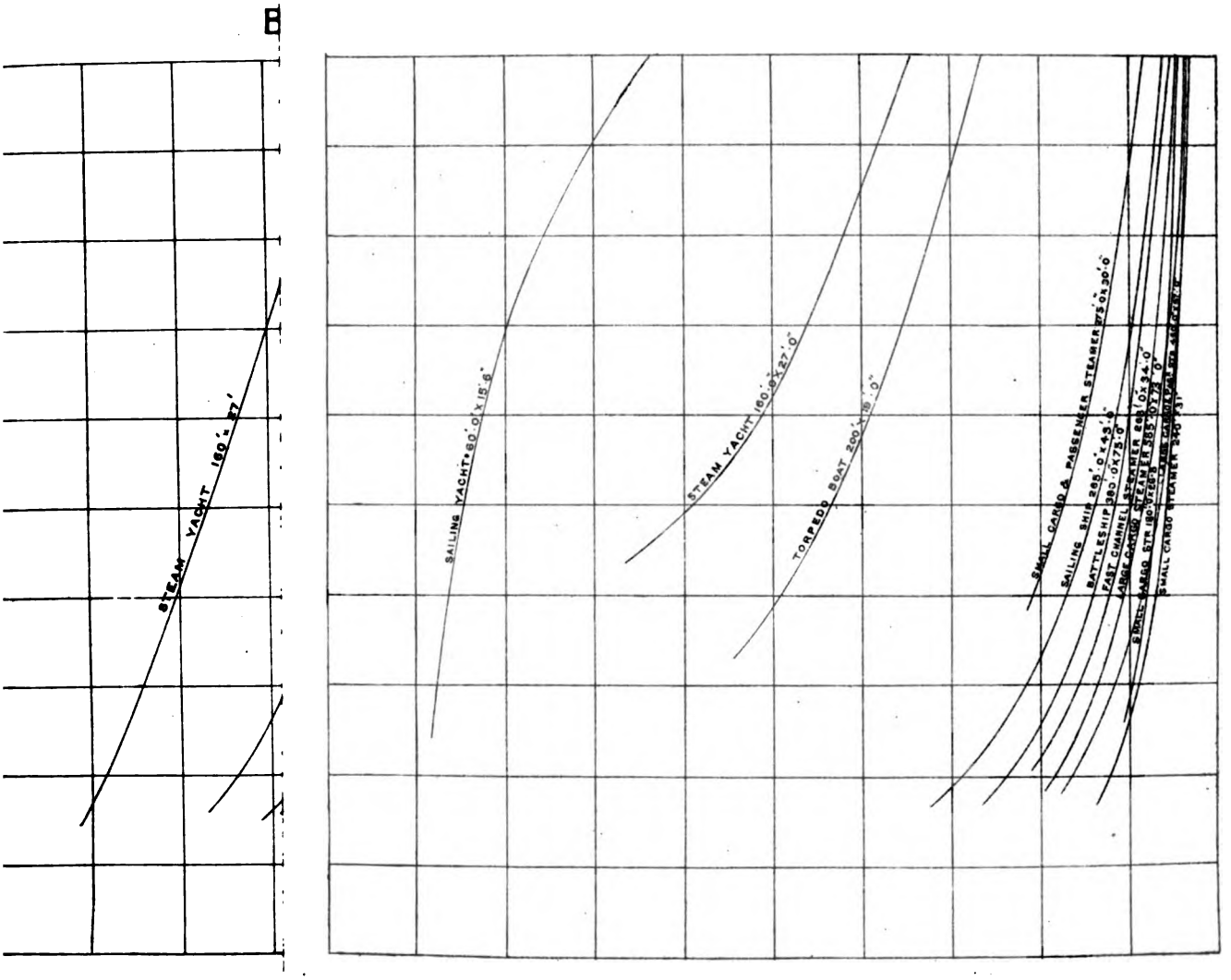
*Calculations.*

# Fig. 11.

## Coefficients of Mid Sections.

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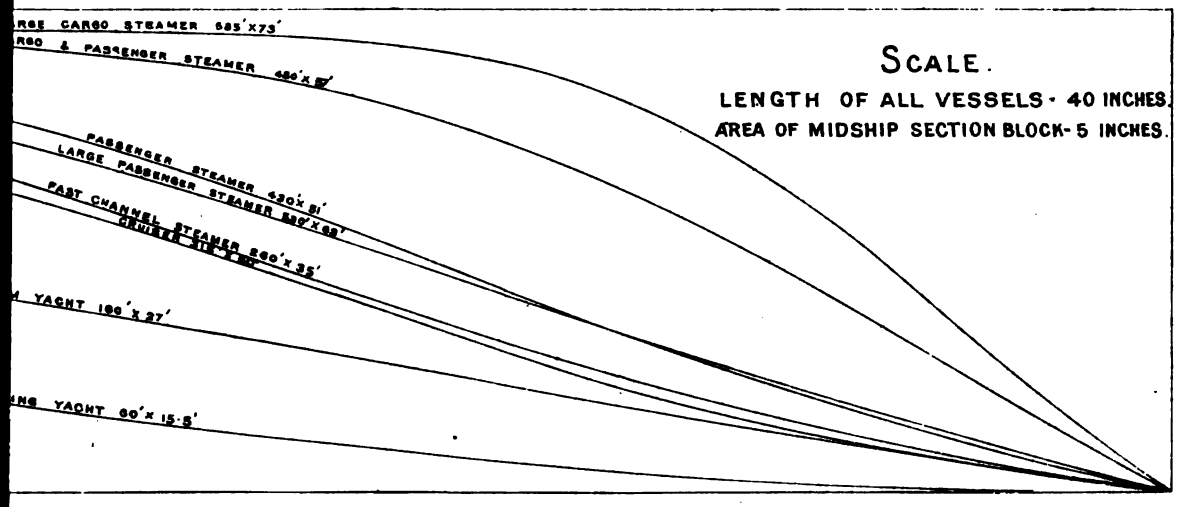


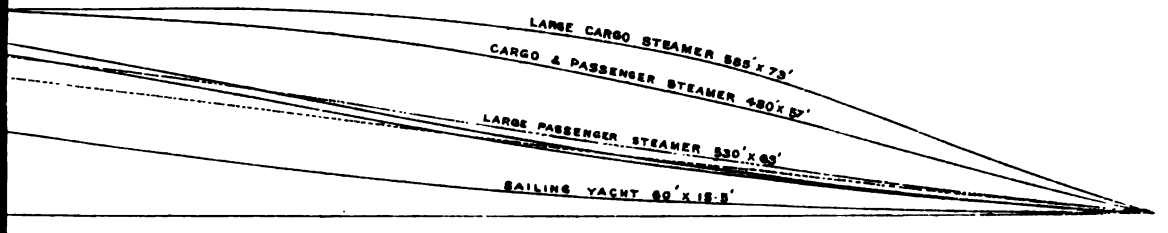


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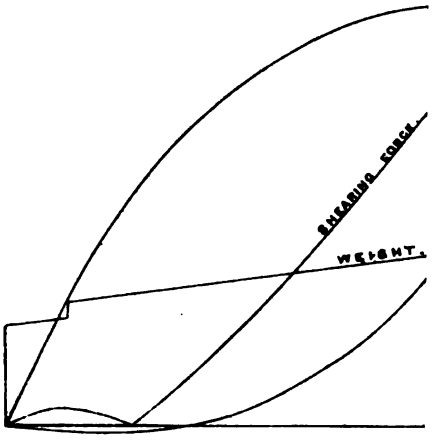


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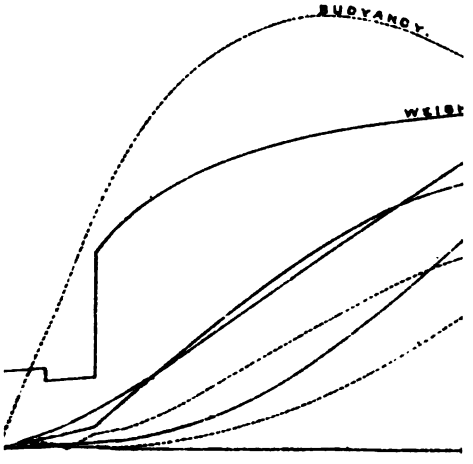


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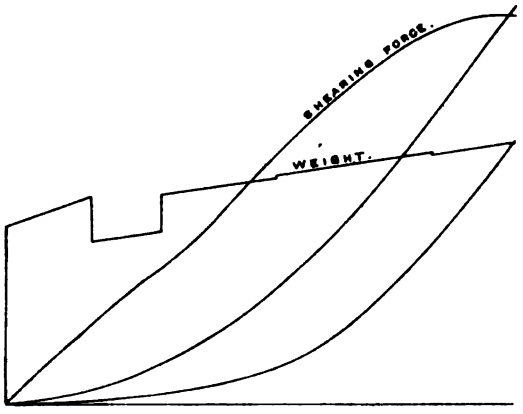


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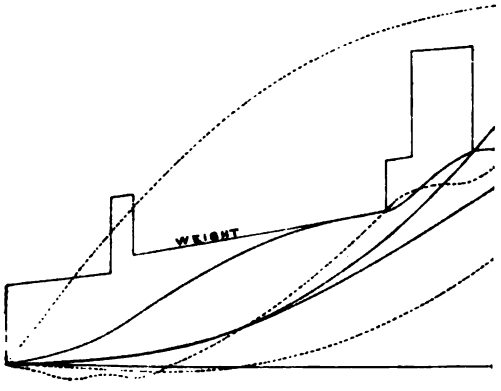


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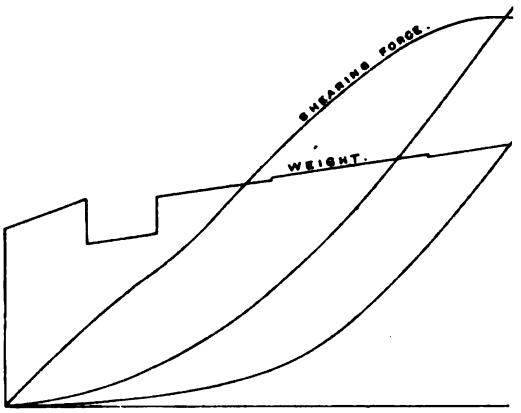
*To Illustrat*

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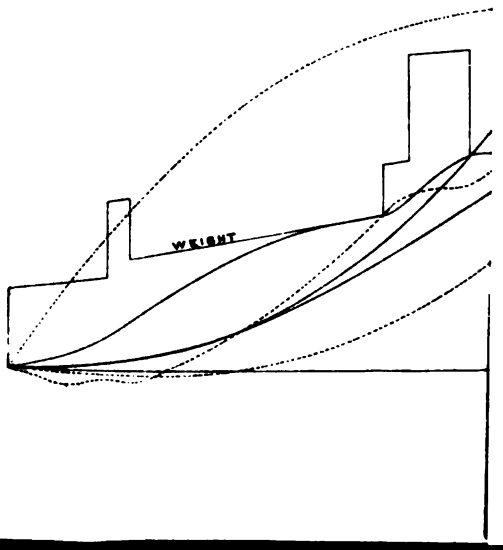


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